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THE ANALYTICAL DESIGN OF A TURBO-JET COMBUSTION CHAMBER

A Thesis Submitted to the Graduate Faculty
of the
University of Minnesota

by
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James B. Verdin

In Partial Fulfillment of the Requirements
for the
Degree of
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I wish to express my thanks to Professors E. J. Robertson, H. A. Hall, T. E. Murphy and J. D. Skersen for their advice and encouragement and to my wife, Muriel, for her help in preparing the manuscript.

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I. SUMMARY

A "can" type combustion chamber was designed using all available proved concepts. A survey of existing designs was made to arrive at a logical design point. The fact that parallel flow would exist within the chamber was noted, and the burner was designed to obtain the required flow with an equal pressure drop across all the parallel paths. The holes for introducing the air into the burner basket were placed in such a manner that the total area into the burner basket, plotted against distance downstream, fell on an arc of a circle. This made the design follow several proved concepts automatically.

The performance was calculated and compared with existing designs. The comparison was quite favorable, with the exception of combustion efficiency. Since the empirical equation for combustion efficiency was obtained from tests under slightly different conditions than the design point, doubt exists as to the validity of the values obtained by its use.

1. The first type of compressor is the reciprocating type. This type of compressor is the most common and is used in a wide variety of applications. It consists of a cylinder in which a piston moves back and forth, compressing the gas. The gas is then discharged into a chamber and the cycle is repeated. This type of compressor is used in a wide variety of applications, including refrigeration, air conditioning, and industrial gas processing.

II. INTRODUCTION

Many combustion chambers for turbo-jet engines have been built and tested; however, they all fall into one of two classifications. The most common type is the tube or "can" combustion chamber. The other is the annular type.

The "can" type of combustion chamber, as its name implies, is a metal cylinder or can with an inner liner or "basket" which separates the air into two parts. The early turbo-jet engines designed by Air Commodore F. Whittle of Great Britain used "can" combustion chambers of the "return flow" type. In this type of chamber, the air passes over the entire length of the combustion zone before entering to be mixed with the fuel and burned. Thus the air is preheated by what would otherwise be waste heat. This advantage is more than overcome by the disadvantage of a high pressure loss. The more modern "can" type combustion chambers are of the "straight through" variety, i.e., there are no turns encountered within the chamber itself.

The annular type of combustion chamber has the distinct advantage of offering less frontal area for a given capacity than a battery of "cans" of equal capacity. The annular combustion chamber has the disadvantage of having to be tested in its entirety, while one "can" of a multi-can battery can be tested. Work is being done at Westinghouse in testing a small segment of an annular chamber and using the results to predict the performance of the complete unit.

The "one" type of composition consists of a single subject or object, which is the only one of its kind in the world. It is a simple, direct, and unambiguous statement of fact. The "two" type of composition consists of two subjects or objects, which are compared or contrasted. It is a more complex statement of fact, requiring the reader to understand the relationship between the two subjects or objects. The "three" type of composition consists of three subjects or objects, which are compared or contrasted. It is a still more complex statement of fact, requiring the reader to understand the relationship between the three subjects or objects. The "four" type of composition consists of four subjects or objects, which are compared or contrasted. It is a very complex statement of fact, requiring the reader to understand the relationship between the four subjects or objects. The "five" type of composition consists of five subjects or objects, which are compared or contrasted. It is an extremely complex statement of fact, requiring the reader to understand the relationship between the five subjects or objects. The "six" type of composition consists of six subjects or objects, which are compared or contrasted. It is a very complex statement of fact, requiring the reader to understand the relationship between the six subjects or objects. The "seven" type of composition consists of seven subjects or objects, which are compared or contrasted. It is an extremely complex statement of fact, requiring the reader to understand the relationship between the seven subjects or objects. The "eight" type of composition consists of eight subjects or objects, which are compared or contrasted. It is a very complex statement of fact, requiring the reader to understand the relationship between the eight subjects or objects. The "nine" type of composition consists of nine subjects or objects, which are compared or contrasted. It is an extremely complex statement of fact, requiring the reader to understand the relationship between the nine subjects or objects. The "ten" type of composition consists of ten subjects or objects, which are compared or contrasted. It is a very complex statement of fact, requiring the reader to understand the relationship between the ten subjects or objects.

No results have been published, but it is expected that this method of testing will prove satisfactory.

Since the "can" type combustion chamber is more easily built and the testing procedures for the chamber are more nearly standardized, it was decided to base the analytical design on this type of combustion chamber.

The volume has been published, and it is expected that this
volume of teaching will prove satisfactory.
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III. PURPOSE OF INVESTIGATION

"Cut and try" has been the method employed in the past in designing combustion chambers for turbo-jet engines. This is a long and costly process, and luck plays a considerable part in obtaining results. It is the purpose of this investigation to design a combustion chamber using analytical methods. Proved concepts will be used where they are available. It is hoped that this procedure will produce a satisfactory chamber with little or no alteration necessary.

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There is a lot of work to be done, and I am sure that the Commission will be able to do it. I am sure that the Commission will be able to do it. I am sure that the Commission will be able to do it.

IV. PROBLEMS INVOLVED IN THE DESIGN OF A COMBUSTION CHAMBER

The function of a turbo-jet combustion chamber is to heat the air from the compressor to the temperature required at the entrance to the turbine. This temperature is usually taken to be 1500°F , this being the maximum temperature at which present day turbine wheels can operate. Sufficient fuel to achieve this temperature gives a fuel-air ratio far too lean for satisfactory combustion. Therefore, the air is divided into primary air and secondary air, the primary air being just sufficient to produce a stoichiometric fuel-air mixture with the required fuel and the secondary air being introduced to cool the products of combustion before reaching the turbine.

The primary air must be introduced into the combustion chamber with enough turbulence to insure good mixing with the fuel. This turbulence also speeds the burning and gives a stable flame.

The secondary air must be brought in with enough large-scale turbulence to complete the burning and to mix thoroughly with the products of combustion. Any stratification due to incomplete mixing will result in layers of air at temperatures higher than the allowable reaching the turbine. Hot spots will be caused, and a failure of the metal may occur.

High turbulence and a long combustion chamber tend to give more complete burning and better mixing of secondary

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has been for satisfactory production. Therefore, the air is
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today just installed in house a solid-state system
which will be required for the secondary air being
introduced to each of the units of production before work
can be done.

The primary aim of the Department is to ensure that children with mental health problems receive the best possible care and support, and that their needs are met in a timely and effective manner.

The following are the results of the investigation:

also have sought testing and better skills of swimming

air and combustion products. Unfortunately, these characteristics are not compatible with an efficient design. High turbulence produces high pressure losses through the combustion chamber. Since compressor efficiencies and pressure ratios are critical, the additional pressure loss which results from high turbulence must be avoided. Therefore, just enough turbulence to complete the mixing should be used. Anything in excess of this, while insurance against turbine hot spots, cuts down the combustion chamber efficiency.

Since turbo-jet engines are used on airplanes, space and weight are at a premium. This precludes the use of an excessively long combustion chamber to complete the mixing. Lengths of three to four diameters are current practice.

The rate at which the secondary air is mixed with the combustion products is of great importance. Too much air introduced too soon may chill the flame front sufficiently to prevent the combustion reaction from going to completion. This, of course, would reduce the combustion efficiency. Reference 1 states that the first secondary air should not be introduced nearer than six to eight inches downstream from the fuel nozzle.

Experiments conducted at the California Institute of Technology show that the shape of the holes used to let the air through the inner liner has an effect on the pressure drop through these holes and also an effect on the mixing of the air with the combustion products. Figures 1 and 2 show the effects of shape on mixing. A thin slot parallel to the

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flow gives the best mixing, a square or circular hole next best, and a slot perpendicular to the flow the worst.

The square hole and the slot are undesirable, however, because high stress concentrations would be produced at the sharp corners.

The round hole has found almost universal acceptance. The use of a bell mouth on these holes reduces the pressure drop across the holes by nearly thirty per cent without affecting the mixing.

These experiments also showed that arranging the holes in lines with one hole directly downstream from the other gave the greatest amount of turbulence for a given pressure drop.

Since there are no facilities available for testing the altitude performance of a combustion chamber, no attempt will be made to take this requirement into consideration.

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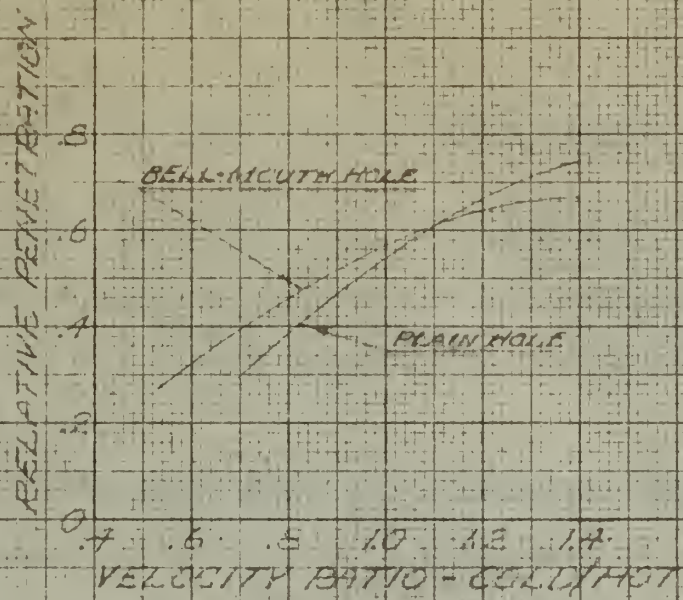


FIGURE 1
EFFECT OF HOLE MOUTH
SHAPE ON MIXING.
(FROM REF. 1)

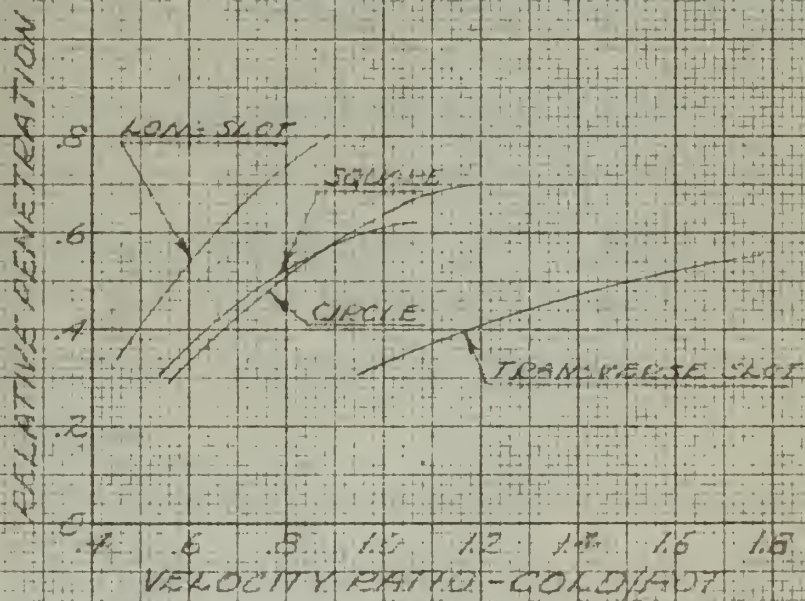


FIGURE 2
EFFECT OF HOLE SHAPE
ON MIXING.
(FROM REF. 1)

V. SELECTION OF THE DESIGN POINT

In order to arrive at a suitable design point, a survey of several current model turbo-jet engines was made. Table I lists the design factors for the combustion chambers of these engines.

The "loading factor" or "combustion intensity factor," as it is sometimes called, is a parameter originated by the British for comparing different combustion chambers. It is defined as follows:

$$I = \frac{q}{PV}$$

where q = heat input in Btu/hr

P = pressure inside chamber in atmospheres

V = volume available for combustion inside basket in ft^3

I = loading factor in Btu/hr- ft^3 -atm.

The heat input is computed by multiplying the fuel flow rate in pounds per hour by the lower heating value of the fuel. The volume available for combustion inside the basket is taken from the fuel injection nozzle to the turbine entrance.

It can readily be seen that the combustion intensity factor is a good indication of the conditions under which combustion takes place. For instance, for a given heat input, the larger the volume available for combustion, the greater will be the amount of air available to absorb the heat released and, consequently, the lower is the intensity

V. RELATION OF THE LINES POINT

In order to arrive at a suitable design point, a set of several hundred random points was generated and the results of the design point for the construction of the design point are shown in Table I.

The design point is a point in the design space which is a point in the design space which is a point in the design space. It is a point in the design space which is a point in the design space. It is a point in the design space which is a point in the design space.

$$x = \frac{1}{\sqrt{2}}$$

where x is the point in the design space.

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factor. Again, for a given heat input and a given volume available for combustion, the lower the pressure, the smaller will be the mass of air and the higher is the temperature rise from combustion; hence, the combustion intensity factor will be higher.

Table I. Design Factors for Combustion Chambers of Current Model Turbo-Jet Engines.*

Engine	N	v_r	I	D	1/D	L/D
TG-180 (J-35)	8	100	3.1×10^6	7.3	3.35	4.80
I-40 (J-33)	14	130	4.6×10^6	5.5	3.60	4.80
Nene	9	80	2.0×10^6	10.0	2.80	2.70
Derwort V	9	80	2.0×10^6	10.0	2.60	3.30
Goblin II	16	100	3.5×10^6	6.0	3.80	4.50
19XB	Annular	-	9.0×10^6	-	-	-
9.5B	Annular	-	10.0×10^6	-	-	-

* I - Loading factor or combustion intensity factor, Btu per hour per cubic foot per atmosphere, based on total volume available for combustion and mixing.

N - Number of combustion chambers.

l - Length in inches from fuel nozzle to end of basket.

L - Length in inches from fuel nozzle to turbine entrance.

v_r - Reference velocity in feet per second, computed using maximum cross-sectional area of combustion chamber.

D - Diameter of basket in inches.

The British have tried to correlate this combustion intensity factor with combustion losses. While no close

Further, the value of α is given by the ratio of the available for combustion, the lower the pressure, the smaller will be the mass of air and the higher is the temperature of the combustion products, the smaller the value of α will be.

Table 1. Values of the coefficient α for different values of the pressure and temperature.

Pressure, MPa	Temperature, °C	α	$\alpha \cdot 10^3$	$\alpha \cdot 10^3$	$\alpha \cdot 10^3$	$\alpha \cdot 10^3$
10-150 (4-15)	0	1.1	1.1	1.1	1.1	1.1
1-10 (0.1-1)	10	1.1	1.1	1.1	1.1	1.1
0.1-1	20	1.1	1.1	1.1	1.1	1.1
0.01-0.1	30	1.1	1.1	1.1	1.1	1.1
0.001-0.01	40	1.1	1.1	1.1	1.1	1.1
0.0001-0.001	50	1.1	1.1	1.1	1.1	1.1
0.00001-0.0001	60	1.1	1.1	1.1	1.1	1.1
0.000001-0.00001	70	1.1	1.1	1.1	1.1	1.1
0.0000001-0.000001	80	1.1	1.1	1.1	1.1	1.1
0.00000001-0.0000001	90	1.1	1.1	1.1	1.1	1.1
0.000000001-0.00000001	100	1.1	1.1	1.1	1.1	1.1

1 - loading factor of combustion intensity factor, the value of the factor is given by the ratio of the available for combustion, the lower the pressure, the smaller will be the mass of air and the higher is the temperature of the combustion products, the smaller the value of α will be.

2 - loading factor of combustion intensity factor, the value of the factor is given by the ratio of the available for combustion, the lower the pressure, the smaller will be the mass of air and the higher is the temperature of the combustion products, the smaller the value of α will be.

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7 - loading factor of combustion intensity factor, the value of the factor is given by the ratio of the available for combustion, the lower the pressure, the smaller will be the mass of air and the higher is the temperature of the combustion products, the smaller the value of α will be.

8 - loading factor of combustion intensity factor, the value of the factor is given by the ratio of the available for combustion, the lower the pressure, the smaller will be the mass of air and the higher is the temperature of the combustion products, the smaller the value of α will be.

9 - loading factor of combustion intensity factor, the value of the factor is given by the ratio of the available for combustion, the lower the pressure, the smaller will be the mass of air and the higher is the temperature of the combustion products, the smaller the value of α will be.

correlation exists between different combustion chambers, curves of individual chambers show that combustion losses increase rapidly with increasing combustion intensity factor. A typical graph of combustion losses vs. combustion intensity factor is presented in Figure 3 for the Lucas B-37 chamber. However, these losses may be inordinately high since, at sea level, values of combustion intensity factor as high as 9×10^6 are possible with a combustion efficiency of 95 per cent (Ref. 2).

The reference velocity, v_r , is another British parameter. It is used to get a rough check on the total losses through the combustion chamber. Reference 3 shows that individual combustion chambers have a pressure loss of from 20 to 30 times the velocity head computed for the reference velocity. Thus it can be seen that a low value of reference velocity should be used in designing a combustion chamber.

Some of the design conditions were dictated by the air source which would be available to test the combustion chamber when built. This source is from the second stage of an Allison V-1710 supercharger and is about three pounds per second at 1.6 atmospheres and 200° F.

As a result of this survey and from consideration of the air source available for testing, the following design point was selected:

Altitude = sea level

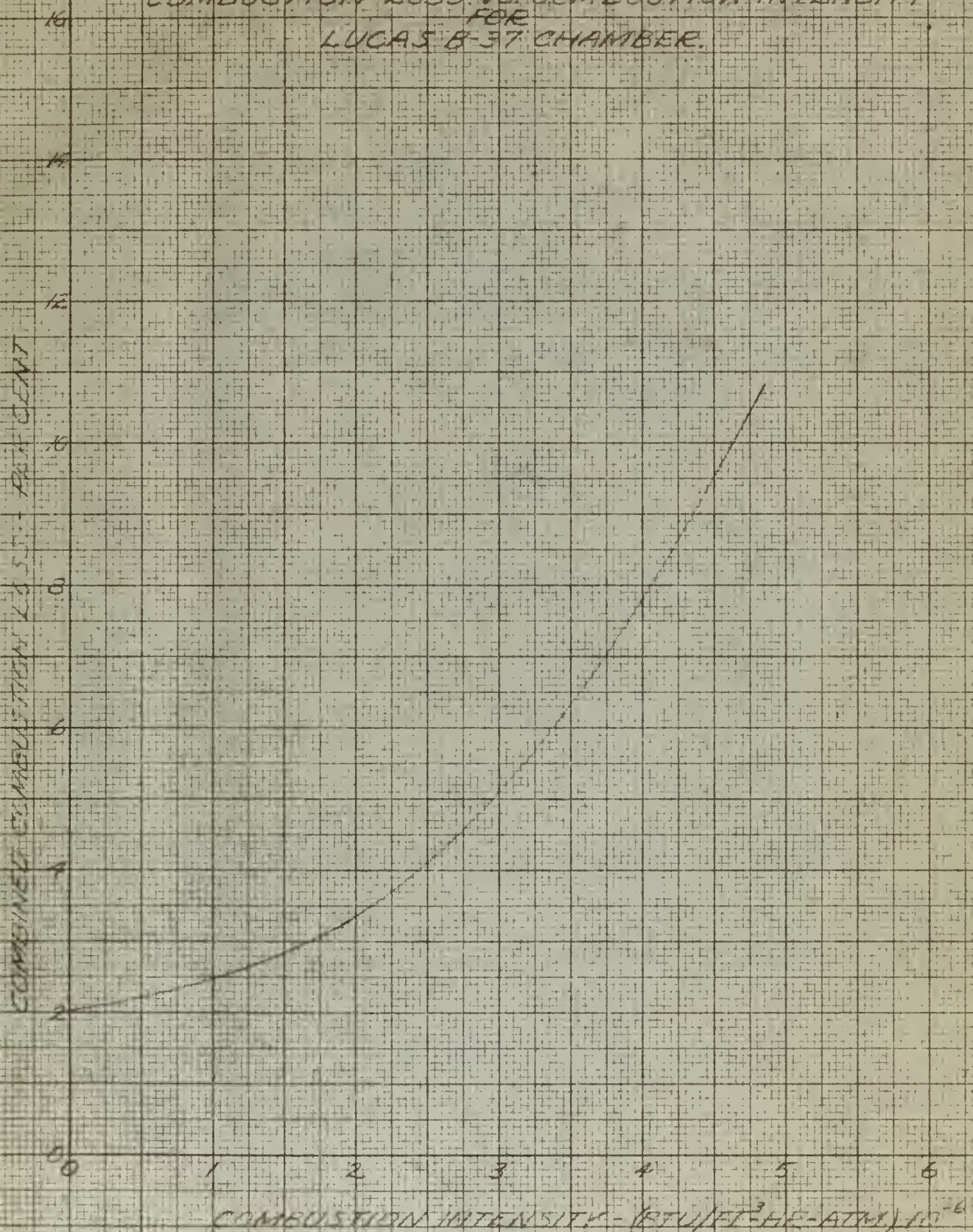
$w = 3 \text{ lb/sec}$

[illegible]

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As a result of this survey and from analysis of the data collected, the following conclusions were drawn:

FIGURE 3
COMBUSTION LOSS VS. COMBUSTION INTENSITY
FOR
LUCAS 8-37 CHAMBER.



$P_{4T} = 1.6$ atmospheres

$T_{4T} = 200^{\circ} F$ or $660^{\circ} R$

$T_{5T} = 1500^{\circ} F$ or $1960^{\circ} R$

$v_r = 100$ ft/sec

$M_r = .05$

$L/D =$ between 3 and 4

$1/D =$ between 2 and 3.

$\gamma_{11} = 1.0$ (assumed)
 $\gamma_{12} = 0.0$ or 0.01
 $\gamma_{22} = 1.000$ or 1.001
 $\gamma_{13} = 0.0$ or 0.01
 $\gamma_{23} = 0.0$
 $\gamma_{33} = 1.0$ or 1.001
 $\gamma_{14} = 0.0$ or 0.01
 $\gamma_{24} = 0.0$ or 0.01
 $\gamma_{34} = 0.0$ or 0.01
 $\gamma_{44} = 1.0$ or 1.001

VI. DESIGN PROCEDURE AND CALCULATIONS

Maximum Diameter of Combustion Chamber

$$v_r = M_r(a)$$

where v_r = reference velocity in ft/sec

M_r = reference Mach number

$$a = \text{speed of sound in ft/sec} = 49\sqrt{T_{4T}}$$

where T_{4T} = temperature in $^{\circ}\text{R}$ at entrance to combustion chamber

$$\begin{aligned} v_r &= .08 \times 49\sqrt{660} \\ &= 100 \text{ ft/sec} \end{aligned}$$

$$\rho = \rho_{sc} \times \frac{P_{4T}}{P_{sc}} \times \frac{T_{sc}}{T_{4T}}$$

where ρ = density of air in lb/ft³

ρ_{sc} = density of air in lb/ft³ at standard conditions

P_{4T} = total pressure in lb/ft² at entrance to combustion chamber

P_{sc} = atmospheric pressure in lb/ft² at standard conditions

T_{sc} = temperature in $^{\circ}\text{R}$ at standard conditions

$$\begin{aligned} \rho &= .07651 \times \frac{2380}{2115} \times \frac{520}{660} \\ &= .0965 \text{ lb/ft}^3 \end{aligned}$$

$$A = \frac{W}{\rho v_r}$$

VI. Other (specify the calculation)

Section 1041(a) of Internal Revenue Code

1041(a)(1)

1041(a)(2) - reference subject to 1041(a)

1041(a)(3) - reference subject to 1041(a)

1041(a)(4) - reference subject to 1041(a)

1041(a)(5) - reference subject to 1041(a)

1041(a)(6) - reference subject to 1041(a)

1041(a)(7) - reference subject to 1041(a)

1041(a)(8) - reference subject to 1041(a)

$$\frac{1041(a)(9)}{1041(a)(10)} = \frac{1041(a)(11)}{1041(a)(12)}$$

1041(a)(13) - reference subject to 1041(a)

1041(a)(14) - reference subject to 1041(a)

1041(a)(15) - reference subject to 1041(a)

1041(a)(16)

1041(a)(17) - reference subject to 1041(a)

1041(a)(18)

1041(a)(19) - reference subject to 1041(a)

$$\frac{1041(a)(20)}{1041(a)(21)} = \frac{1041(a)(22)}{1041(a)(23)}$$

1041(a)(24)

$$\frac{1041(a)(25)}{1041(a)(26)}$$

where A = area of combustion chamber in ft^2

w = air flow in lb/sec

$$A = \frac{3}{.0965 \times 100}$$

$$= .29 \text{ ft}^2$$

$$D = \sqrt{\frac{A}{.785}}$$

where D_{max} = maximum diameter of combustion chamber in ft

$$D_{\text{max}} = \sqrt{\frac{.29}{.785}}$$

$$= .61 \text{ ft} = 7.33 \text{ in}$$

Fuel Required

$$c_{\text{pave}} = \frac{c_{p4} + c_{p5}}{2}$$

where c_{pave} = average specific heat at constant pressure
in $\text{Btu/lb air} - ^\circ\text{R}$

c_{p4} = specific heat at constant pressure in
 $\text{Btu/lb air} - ^\circ\text{R}$ at entrance to combustion
chamber

c_{p5} = specific heat at constant pressure in Btu/lb
air - $^\circ\text{R}$ at exit from combustion chamber

$$c_{\text{pave}} = \frac{.2403 + .277}{2}$$

$$= .256 \text{ Btu/lb air} - ^\circ\text{R}$$

$$q = c_{\text{pave}} (T_{5T} - T_{4T})$$

where q = heat required in Btu/lb air

T_{5T} = total temperature at combustion chamber exit in
 $^\circ\text{R}$

where Δ is the difference between the two

Δ is the difference between the two

$$\Delta = \frac{1}{2} \left(\frac{1}{\rho_1} + \frac{1}{\rho_2} \right)$$

Δ is the difference between the two

$$\Delta = \frac{1}{2} \left(\frac{1}{\rho_1} + \frac{1}{\rho_2} \right)$$

where Δ is the difference between the two

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$$\Delta = \frac{1}{2} \left(\frac{1}{\rho_1} + \frac{1}{\rho_2} \right)$$

Δ is the difference between the two

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Δ is the difference between the two

Δ is the difference between the two

$$q = .256(1960 - 660)$$

$$= 343 \text{ Btu/lb air}$$

$$w_f = \frac{q}{\text{L.H.V.}}$$

where w_f = fuel required in lb/sec

L.H.V. = lower heating value of fuel in Btu/lb

$$w_f = \frac{343 \times 3}{18500}$$

$$= .056 \text{ lb fuel/sec}$$

Primary Air Required

In order to obtain sufficient combustion in the primary zone, there should be a stoichiometric mixture of fuel and air in this zone. Thus, the fuel-air ratio must be .067.

$$w = \frac{w_f}{.067}$$

where w = weight of primary air required in lb/sec

$$w = \frac{.056}{.067}$$

$$= .85 \text{ lb/sec}$$

Volume Available for Combustion

$$V = \frac{q}{F_i}$$

where V = volume available for combustion in ft^3

q = heat input in Btu/hr

$$p = 1.013 \times 10^5 \text{ N/m}^2$$

$$p = 1.013 \times 10^5 \text{ N/m}^2$$

$$\frac{p}{\rho g h} = 10$$

where p = fluid pressure in N/m²

ρ = fluid density in kg/m³

$$\frac{1.013 \times 10^5}{1000 \times 9.81} = 10$$

$$h = 10.33 \text{ m}$$

PROBLEM 10.10

In order to obtain sufficient ventilation in the

factory room, there should be a mechanical system of

draw and air in this room. Find the flow rate in m³/s

in 1000.

$$\frac{p}{\rho g h} = 10$$

where p = fluid pressure in N/m²

$$\frac{p}{\rho g h} = 10$$

$$h = 10.33 \text{ m}$$

PROBLEM 10.11

$$\frac{p}{\rho g h} = 10$$

where p = fluid pressure in N/m²

$$h = 10.33 \text{ m}$$

P = pressure inside chamber in atmospheres

I = combustion intensity factor in Btu/hr-ft³-atm

$$V = \frac{.056 \times 3600 \times 16500}{1.6 \times 8 \times 10^6}$$

$$= .29 \text{ ft}^3$$

Configuration of Burner Basket

As shown in Figure 4, the burner is made up of three sections---hemisphere or dome, cylinder, and tail cone.

Thus,

$$V_T = V_H + V_C + V_{TC}$$

where V_T = total volume in ft³ = .29 ft³

V_H = volume of hemisphere in ft³

V_C = volume of cylinder in ft³

V_{TC} = volume of tail cone in ft³

Then,

$$V_T = \frac{\pi D^3}{12} + \frac{\pi D^2 l_C}{4} + \frac{h}{3} \left(\frac{\pi D^2}{4} + \frac{\pi d^2}{4} + \sqrt{\frac{\pi D^2}{4} \times \frac{\pi d^2}{4}} \right)$$

where D = diameter of cylinder in ft

d = smaller diameter of tail cone in ft

l_C = length of cylinder in ft

h = length of tail cone in ft

Assuming that D = 6 in = 1/2 ft, d = 4 in = 1/3 ft,
and h = 5 in = 5/12 ft:

$$.29 = \frac{.785}{4} l_C + \frac{.262}{3} + \frac{5}{24} \left(\frac{.785}{4} + \frac{.785}{9} + \sqrt{\frac{.785}{4} \times \frac{.785}{9}} \right)$$

1. A certain number of men can do a piece of work in 10 days. If 10 more men are added, the work can be done in 8 days. How many men were there at first?

$$\frac{10x}{10} = \frac{(x+10)8}{8}$$

$$x = 40$$

PROBLEMS ON WORK

1. A certain number of men can do a piece of work in 10 days. If 10 more men are added, the work can be done in 8 days. How many men were there at first?

$$x = 40$$

2. A certain number of men can do a piece of work in 10 days. If 10 more men are added, the work can be done in 8 days. How many men were there at first?

3. A certain number of men can do a piece of work in 10 days. If 10 more men are added, the work can be done in 8 days. How many men were there at first?

4. A certain number of men can do a piece of work in 10 days. If 10 more men are added, the work can be done in 8 days. How many men were there at first?

5. A certain number of men can do a piece of work in 10 days. If 10 more men are added, the work can be done in 8 days. How many men were there at first?

$$\left(\frac{10x}{10} + \frac{10x}{8} \right) = \frac{10x}{10} + \frac{10x}{8}$$

6. A certain number of men can do a piece of work in 10 days. If 10 more men are added, the work can be done in 8 days. How many men were there at first?

7. A certain number of men can do a piece of work in 10 days. If 10 more men are added, the work can be done in 8 days. How many men were there at first?

8. A certain number of men can do a piece of work in 10 days. If 10 more men are added, the work can be done in 8 days. How many men were there at first?

9. A certain number of men can do a piece of work in 10 days. If 10 more men are added, the work can be done in 8 days. How many men were there at first?

10. A certain number of men can do a piece of work in 10 days. If 10 more men are added, the work can be done in 8 days. How many men were there at first?

$$x = 40$$

$$\left(\frac{10x}{10} + \frac{10x}{8} \right) = \frac{10x}{10} + \frac{10x}{8}$$

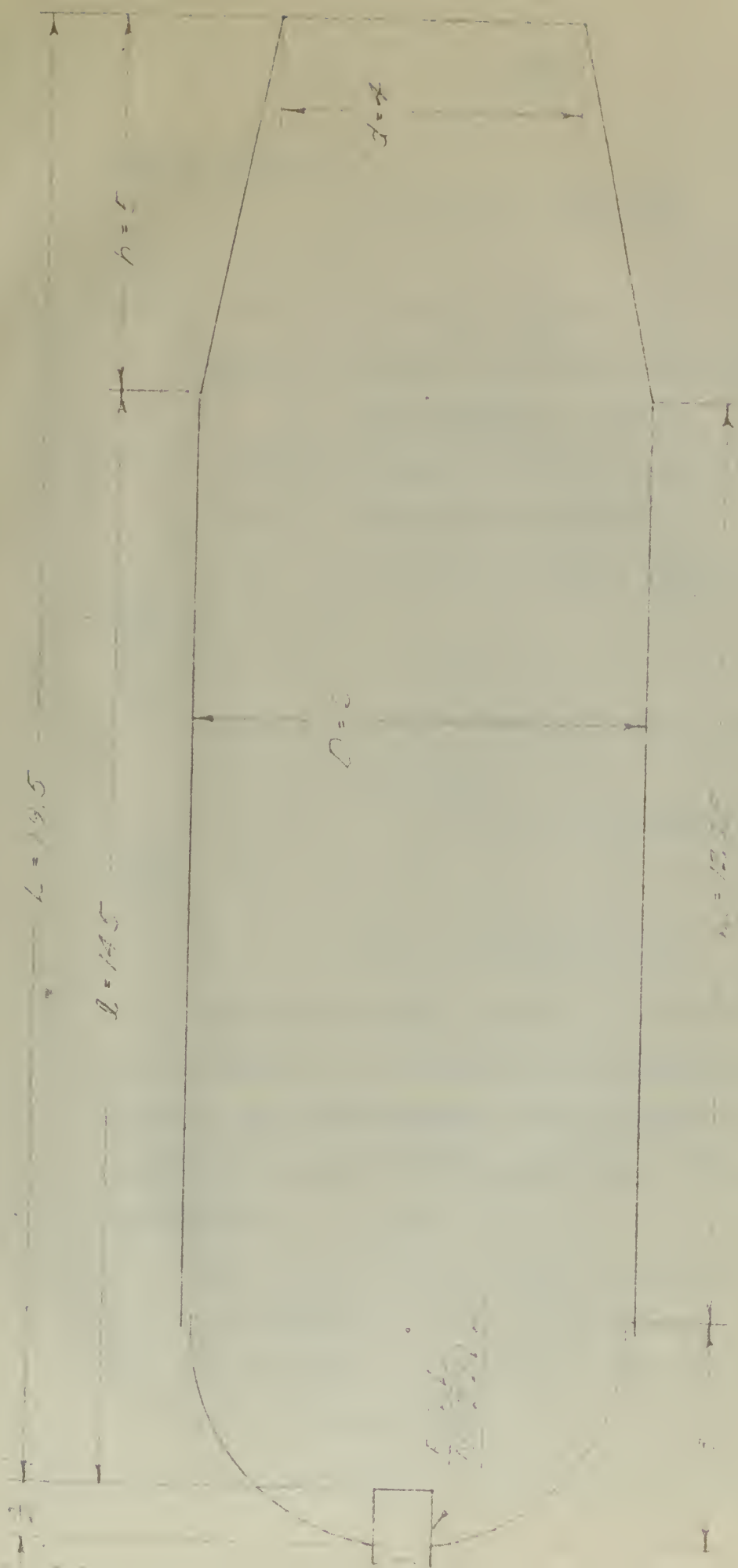


FIGURE 4 SKETCH OF BURNER BASKET, DOME, AND TAIL CONE.

Solving for l_C ,

$$l_C = 1.02 \text{ ft} = 12.25 \text{ in}$$

Then,

$$l = l_C + 2.25$$

where l = length from fuel nozzle to downstream end
of cylinder in inches

2.25 = length from fuel nozzle to upstream end of
cylinder in inches

$$l = 12.25 + 2.25 = 14.5 \text{ in}$$

Also,

$$L = l + h$$

where L = length from fuel nozzle to end of tail cone
in inches

$$L = 14.5 + 5 = 19.5 \text{ in}$$

Then,

$$\frac{l}{D} = 2.42 \text{ and } \frac{L}{D} = 3.25$$

Since the values of l/D and L/D are between the desired values of 2 and 3 and 2 to 4 respectively, the dimensions of the burner basket were selected as assumed and calculated in the foregoing procedure and may be summarized as follows:

$$D = 6 \text{ in}$$

$$l_C = 12.25 \text{ in}$$

$$d = 4 \text{ in}$$

$$l = 14.5 \text{ in}$$

$$h = 5 \text{ in}$$

$$L = 19.5 \text{ in}$$

calculated for 10.

$$10 = 1.00 \text{ ft} = 12.00 \text{ in}$$

Then,

$$1 = 12 = 0.80$$

where $1 =$ length from level outside to basement and

or equivalent in inches

$$2.51 = \text{length from level outside to basement and 10}$$

calculated in inches

$$1 = 12.00 = 0.80 = 14.8 \text{ in}$$

Then,

$$1 = 1 = 0$$

where $1 =$ length from level outside to end of full room

in inches

$$1 = 12.00 + 0 + 10.0 = 22$$

Then,

$$\frac{1}{2} = 0.50 \text{ and } \frac{1}{2} = 0.50$$

Since the values of $1/2$ and $1/2$ are between the 50-

and values of 0 and 1 to 0 respectively, the above-

values of the number of units were selected as required and

calculated in the following procedure and may be summarized

as follows:

$$10 = 0 \text{ in} \quad 10 = 12.00 \text{ in}$$

$$0 = 0 \text{ in} \quad 1 = 12.00 \text{ in}$$

$$0 = 0 \text{ in} \quad 1 = 12.00 \text{ in}$$

Design of Primary Zone

The problem of introducing the primary air with the proper turbulence is the most important single factor in the design of a combustion chamber, according to A. J. Nerad (Ref. 4). When done properly, a smooth, continuous ignition process occurs in the dome end of the combustion chamber.

Best results have been obtained when everything possible has been done to produce a strong reverse flow in the primary zone. This reverse or "back-flow" produces an "ignition eddy" or "tore," as it is referred to in Nerad's paper.

To establish the existence of this "tore" beyond any doubt, various tests have been carried out with pressure probes, along with other tests such as injecting the fuel into the first ring of holes instead of at the nozzle. These tests have proved that the flow is essentially that which is shown in Figure 5.

Looking at the end view of Figure 5, it is obvious that the eight strong jets of air impacting at the center must produce a substantial axial flow both into and out of the plane of the paper. The axial flow which goes forward is deflected outward by the dome and emerges in the relatively unobstructed areas between the rows of air inlets. This air has mixed with fuel and been ignited and now is "hot burning gases." These gases pass in close proximity to the incoming cold air and part is entrained by it, re-

History of the ...

The history of ...

... is the most important ...

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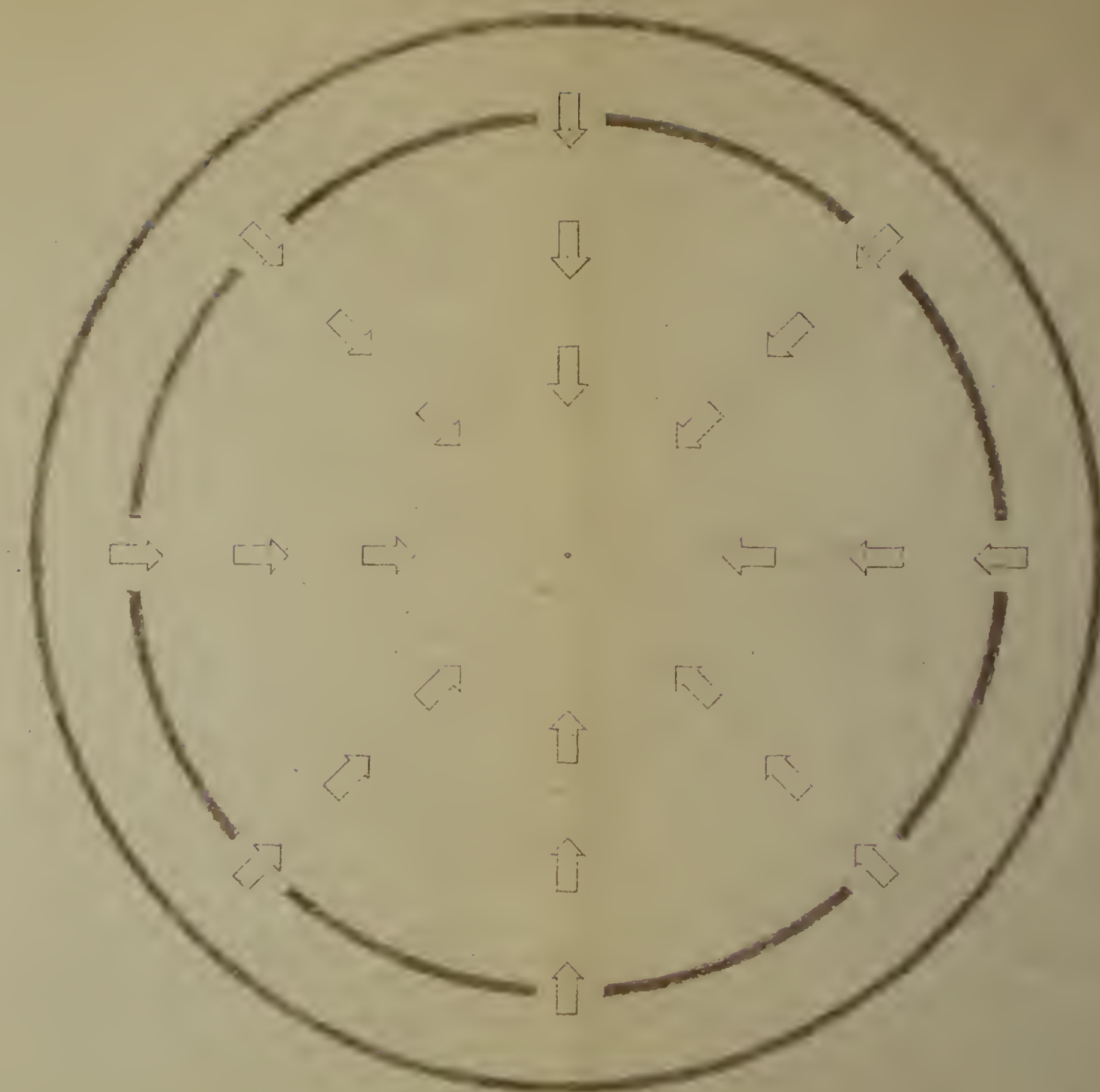
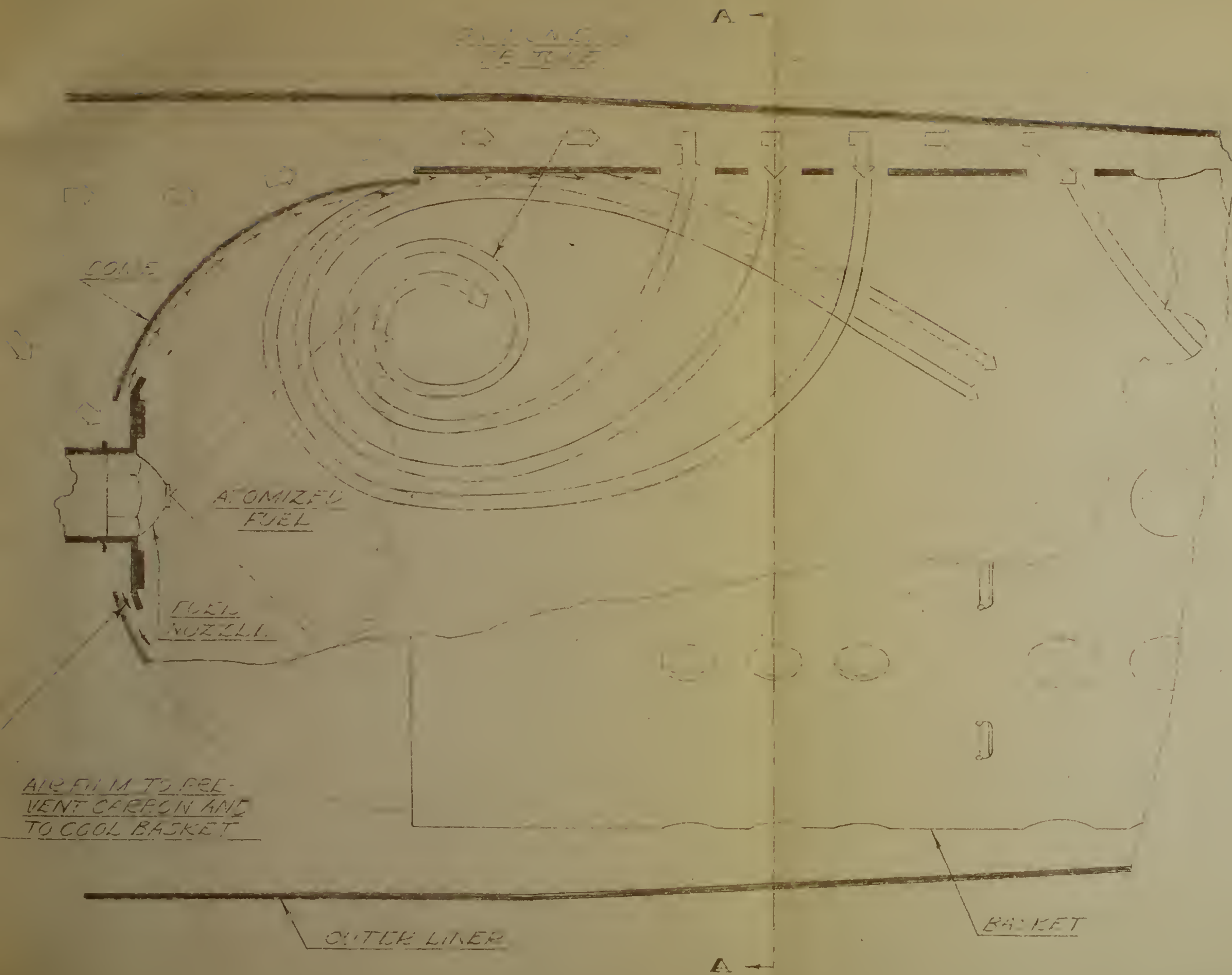
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SECTION A-A

FIGURE 4. SCHEMATIC DIAGRAM OF COMBUSTION CHAMBER SHOWING FLOW OF AIR.

sulting in extremely fast preheating of the incoming combustion air. This has a very beneficial effect on good ignition and smooth combustion.

Since it is desirable to have this "back-flow" as strong as possible, all the primary air, except a small amount necessary for cooling and carbon prevention, will be introduced through holes in the basket downstream from the fuel nozzle.

The flow through the combustion chamber is parallel, i.e., the mass of air which enters the combustion chamber through any hole or slot in the burner basket does not in turn pass through any other hole or slot on its way to the turbine, and, likewise, the air which travels through the annular path in the tail cone has not previously passed through any hole or slot in the basket. Since the flow is parallel, the friction pressure drop through any part of the burner must equal that through any other part. This fact was used in designing the combustion chamber.

The design of the primary zone was completed as follows:

First, the weight flow through a ring of eight 1/2 inch primary holes located 4-1/2 inches from the nozzle was determined approximately by the formula:

$$w = \rho A_2 v C_o$$

where w = air flow through the ring of holes in
lb/sec

A_2 = area of the holes in ft²

limited and much smaller.

44 "Continued" this case is scheduled at 11 noon

I have a friend who has been very ill, and I am
very anxious to see him, but I cannot go now.
I will write to you again soon.

Your friend,
John Doe

[illegible]

the having to the contrary and was completed as follows:

These primary data suggest a 0.1% increase from the results
 of the weight loss studies a year or more later.

200

...and the ...

¹¹ 93 mil. 1942-44. 1944-45. 1945-46. 1946-47. 1947-48. 1948-49. 1949-50. 1950-51. 1951-52. 1952-53. 1953-54. 1954-55. 1955-56. 1956-57. 1957-58. 1958-59. 1959-60. 1960-61. 1961-62. 1962-63. 1963-64. 1964-65. 1965-66. 1966-67. 1967-68. 1968-69. 1969-70. 1970-71. 1971-72. 1972-73. 1973-74. 1974-75. 1975-76. 1976-77. 1977-78. 1978-79. 1979-80. 1980-81. 1981-82. 1982-83. 1983-84. 1984-85. 1985-86. 1986-87. 1987-88. 1988-89. 1989-90. 1990-91. 1991-92. 1992-93. 1993-94. 1994-95. 1995-96. 1996-97. 1997-98. 1998-99. 1999-00. 2000-01. 2001-02. 2002-03. 2003-04. 2004-05. 2005-06. 2006-07. 2007-08. 2008-09. 2009-10. 2010-11. 2011-12. 2012-13. 2013-14. 2014-15. 2015-16. 2016-17. 2017-18. 2018-19. 2019-20. 2020-21. 2021-22. 2022-23. 2023-24. 2024-25. 2025-26. 2026-27. 2027-28. 2028-29. 2029-30. 2030-31. 2031-32. 2032-33. 2033-34. 2034-35. 2035-36. 2036-37. 2037-38. 2038-39. 2039-40. 2040-41. 2041-42. 2042-43. 2043-44. 2044-45. 2045-46. 2046-47. 2047-48. 2048-49. 2049-50. 2050-51. 2051-52. 2052-53. 2053-54. 2054-55. 2055-56. 2056-57. 2057-58. 2058-59. 2059-60. 2060-61. 2061-62. 2062-63. 2063-64. 2064-65. 2065-66. 2066-67. 2067-68. 2068-69. 2069-70. 2070-71. 2071-72. 2072-73. 2073-74. 2074-75. 2075-76. 2076-77. 2077-78. 2078-79. 2079-80. 2080-81. 2081-82. 2082-83. 2083-84. 2084-85. 2085-86. 2086-87. 2087-88. 2088-89. 2089-90. 2090-91. 2091-92. 2092-93. 2093-94. 2094-95. 2095-96. 2096-97. 2097-98. 2098-99. 2099-00. 2100-01. 2101-02. 2102-03. 2103-04. 2104-05. 2105-06. 2106-07. 2107-08. 2108-09. 2109-10. 2110-11. 2111-12. 2112-13. 2113-14. 2114-15. 2115-16. 2116-17. 2117-18. 2118-19. 2119-20. 2120-21. 2121-22. 2122-23. 2123-24. 2124-25. 2125-26. 2126-27. 2127-28. 2128-29. 2129-30. 2130-31. 2131-32. 2132-33. 2133-34. 2134-35. 2135-36. 2136-37. 2137-38. 2138-39. 2139-40. 2140-41. 2141-42. 2142-43. 2143-44. 2144-45. 2145-46. 2146-47. 2147-48. 2148-49. 2149-50. 2150-51. 2151-52. 2152-53. 2153-54. 2154-55. 2155-56. 2156-57. 2157-58. 2158-59. 2159-60. 2160-61. 2161-62. 2162-63. 2163-64. 2164-65. 2165-66. 2166-67. 2167-68. 2168-69. 2169-70. 2170-71. 2171-72. 2172-73. 2173-74. 2174-75. 2175-76. 2176-77. 2177-78. 2178-79. 2179-80. 2180-81. 2181-82. 2182-83. 2183-84. 2184-85. 2185-86. 2186-87. 2187-88. 2188-89. 2189-90. 2190-91. 2191-92. 2192-93. 2193-94. 2194-95. 2195-96. 2196-97. 2197-98. 2198-99. 2199-00. 2200-01. 2201-02. 2202-03. 2203-04. 2204-05. 2205-06. 2206-07. 2207-08. 2208-09. 2209-10. 2210-11. 2211-12. 2212-13. 2213-14. 2214-15. 2215-16. 2216-17. 2217-18. 2218-19. 2219-20. 2220-21. 2221-22. 2222-23. 2223-24. 2224-25. 2225-26. 2226-27. 2227-28. 2228-29. 2229-30. 2230-31. 2231-32. 2232-33. 2233-34. 2234-35. 2235-36. 2236-37. 2237-38. 2238-39. 2239-40. 2240-41. 2241-42. 2242-43. 2243-44. 2244-45. 2245-46. 2246-47. 2247-48. 2248-49. 2249-50. 2250-51. 2251-52. 2252-53. 2253-54. 2254-55. 2255-56. 2256-57. 2257-58. 2258-59. 2259-60. 2260-61. 2261-62. 2262-63. 2263-64. 2264-65. 2265-66. 2266-67. 2267-68. 2268-69. 2269-70. 2270-71. 2271-72. 2272-73. 2273-74. 2274-75. 2275-76. 2276-77. 2277-78. 2278-79. 2279-80. 2280-81. 2281-82. 2282-83. 2283-84. 2284-85. 2285-86. 2286-87. 2287-88. 2288-89. 2289-90. 2290-91. 2291-92. 2292-93. 2293-94. 2294-95. 2295-96. 2296-97. 2297-98. 2298-99. 2299-00. 2300-01. 2301-02. 2302-03. 2303-04. 2304-05. 2305-06. 2306-07. 2307-08. 2308-09. 2309-10. 2310-11. 2311-12. 2312-13. 2313-14. 2314-15. 2315-16. 2316-17. 2317-18. 2318-19. 2319-20. 2320-21. 2321-22. 2322-23. 2323-24. 2324-25. 2325-26. 2326-27. 2327-28. 2328-29. 2329-30. 2330-31. 2331-32. 2332-33. 2333-34. 2334-35. 2335-36. 2336-37. 2337-38. 2338-39. 2339-40. 2340-41. 2341-42. 2342-43. 2343-44. 2344-45. 2345-46. 2346-47. 2347-48. 2348-49. 2349-50. 2350-51. 2351-52. 2352-53. 2353-54. 2354-55. 2355-56. 2356-57. 2357-58. 2358-59. 2359-60. 2360-61. 2361-62. 2362-63. 2363-64. 2364-65. 2365-66. 2366-67. 2367-68. 2368-69. 2369-70. 2370-71. 2371-72. 2372-73. 2373-74. 2374-75. 2375-76. 2376-77. 2377-78. 2378-79. 2379-80. 2380-81. 2381-82. 2382-83. 2383-84. 2384-85. 2385-86. 2386-87. 2387-88. 2388-89. 2389-90. 2390-91. 2391-92. 2392-93. 2393-94. 2394-95. 2395-96. 2396-

ρ = density of air in lb/ft³

v = velocity of air through holes in ft/sec and is taken as equal to that in the clearance area between the liner and the burner basket

C_o = orifice coefficient

C_o , the orifice coefficient, was obtained from a graph (Ref. 5) after computing Reynold's number and the diameter ratio as follows:

$$A_1 = \frac{\pi}{4}(D_{od}^2 - D_{id}^2)$$

Where A_1 = clearance area between basket and burner shell at the location of the primary holes in ft²

D_{od} = outside diameter of clearance area in ft

D_{id} = inside diameter of clearance area in ft

$$A_1 = .785 \left[\left(\frac{7.25}{12} \right)^2 - \left(\frac{6.1}{12} \right)^2 \right]$$

$$= .089 \text{ ft}^2 = 12.8 \text{ in}^2$$

$$D_1 = \sqrt{\frac{A_1}{\pi/4}}$$

Where D_1 = diameter in ft of a pipe with area equivalent to A_1

$$D_1 = \sqrt{\frac{.089}{.785}}$$

$$= .336 \text{ ft} = 4.04 \text{ in}$$

At this point, it is assumed that .25 lb air/sec will have been used previously for cooling and carbon prevention.

9 density of air in layer
x a velocity of air through tubes in layer and in
tubes on equal to that in the narrow zone be-
tween the floor and the narrow channel

2. a static coefficient
2.1 the velocity coefficient, and obtained from a
graph (Fig. 1) after coefficient $\alpha_{2.1}$ is found and the
dynamic factor is obtained

$$K_1 = \frac{\pi}{4} \left(\frac{d}{D} \right)^4 \cdot \frac{1}{\alpha_{2.1}}$$

where K_1 is a coefficient that depends on the diameter of the
tubes and the velocity of the flow
where $\alpha_{2.1}$ is

$K_{1.2}$ = velocity coefficient of diameter ratio in 1.2
 $K_{1.3}$ = velocity coefficient of diameter ratio in 1.3

$$K_1 = \frac{1}{\alpha_{2.1}} \left(\frac{d}{D} \right)^4 \cdot \frac{1}{\alpha_{2.1}}$$

$$K_1 = 0.001 \cdot \frac{1}{\alpha_{2.1}} \cdot \frac{1}{D^4}$$

$$K_1 = \frac{1}{\alpha_{2.1}} \cdot \frac{1}{D^4}$$

where K_1 is a coefficient in 1.2 of a pipe with water
equivalent in 1.2

$$K_1 = \frac{1}{\alpha_{2.1}} \cdot \frac{1}{D^4}$$

$$K_1 = 0.001 \cdot \frac{1}{\alpha_{2.1}} \cdot \frac{1}{D^4}$$

at this point, it is assumed that $\alpha_{2.1}$ is 1.0
and that the coefficient for velocity and dynamic factor

$$v_1 = \frac{w_1}{\rho_{a1}}$$

where v_1 = velocity in ft/sec through clearance area

w_1 = air flow in clearance area in lb/sec

$$v_1 = \frac{3 - .25}{.0965 \times .089} = 320 \text{ ft/sec}$$

By Sutherland's formula:

$$\mu = \mu_r \left(\frac{T_r + K}{T + K} \right) \left(\frac{T}{T_r} \right)^{3/2}$$

where μ = viscosity of the air in poises

$$\mu_r = 170.9 \times 10^{-6} \text{ poises}$$

$$T_r = 273^\circ \text{ K}$$

$$K = 120$$

$$T = \text{temperature of air} = 366.4^\circ \text{ K}$$

$$= 170.9 \times 10^{-6} \left(\frac{293}{486.4} \right) \left(\frac{366.4}{273} \right)^{3/2}$$

$$= 2.145 \times 10^{-4} \text{ poises}$$

$$= 2.145 \times 10^{-4} \times 2.09 \times 10^{-3} \text{ slugs/ft-sec}$$

Then,

$$NR = \frac{E_1 v_1 \rho}{\mu g}$$

where NR = Reynold's number

$$g = \text{gravitational constant} = 32.2 \text{ ft/sec}^2$$

$$\frac{1}{p} = \frac{1}{p_0} + \frac{1}{p_1}$$

where p is the probability of a particle being in a certain state at a certain time, p_0 is the probability of a particle being in a certain state at a certain time, and p_1 is the probability of a particle being in a certain state at a certain time.

$$p = \frac{1}{1 + \frac{1}{p_0} + \frac{1}{p_1}}$$

where p is the probability of a particle being in a certain state at a certain time, p_0 is the probability of a particle being in a certain state at a certain time, and p_1 is the probability of a particle being in a certain state at a certain time.

$$p = \frac{1}{1 + \frac{1}{p_0} + \frac{1}{p_1}}$$

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where

where p is the probability of a particle being in a certain state at a certain time, p_0 is the probability of a particle being in a certain state at a certain time, and p_1 is the probability of a particle being in a certain state at a certain time.

$$p = \frac{1}{1 + \frac{1}{p_0} + \frac{1}{p_1}}$$

$$NR = \frac{.336 \times 320 \times .0965}{.45 \times 10^{-6} \times 32.2}$$

$$= .716 \times 10^6$$

$$A_2 = \frac{\pi N_h D_h^2}{4}$$

where N_h = number of holes in ring

D_h = diameter of each hole in ft

$$A_2 = .785 \times 8 \times \left(\frac{1/2}{12}\right)^2$$

$$= .011 \text{ ft}^2 = 1.56 \text{ in}^2$$

$$\frac{D_2}{D_1} = \sqrt{\frac{A_2}{A_1}} = \sqrt{\frac{1.56}{12.8}} = .35$$

Then, with these values of NR and D_2/D_1 and the chart of Reference 5, we find $C_o = .61$.

Thus,

$$w = .0965 \times .011 \times 320 \times .61$$

$$= .2 \text{ lb/sec}$$

$$Q = \frac{w}{\rho}$$

where Q = air flow through ring of holes in ft^3/sec

$$Q = \frac{.2}{.0965} = 2.07 \text{ ft}^3/\text{sec}$$

The friction pressure drop was then calculated by the following formula from Reference 6:

$$\frac{1000 \times 1000 \times 1000}{1000 \times 1000 \times 1000} = 1$$

$$1000 \times 1000 = 1000000$$

$$\frac{1000}{1000} = 1$$

above $\frac{1}{2}$ = number of units in 10
 $\frac{1}{2}$ = number of units in 10

$$\left(\frac{1}{2}\right)^2 = \frac{1}{4}$$

$$1000 \times 1000 = 1000000$$

$$1000 \times 1000 = 1000000$$

Then, after these units of 10 and 10^2 and the
 units of 10^3 and 10^4 = 10.

Thus,

$$1000 \times 1000 = 1000000$$

$$1000 \times 1000 = 1000000$$

$$1000 \times 1000 = 1000000$$

where $\frac{1}{2}$ = the number of units in 10^2 and

$$\frac{1000}{1000} = 1$$

The following figures are also indicated by
 the following figures: 1000000

$$\Delta P = \frac{\rho}{2g} \left[\frac{2}{A_2 C_o} \sqrt{1 - \left(\frac{A_2}{A_1} \right)^2} \right]^2$$

where ΔP = friction pressure drop in lb/ft²

$$\begin{aligned} \Delta P &= \frac{.0965}{2 \times 32.2} \left[\frac{2.07}{.011 \times .61} \sqrt{1 - \left(\frac{.011}{.089} \right)^2} \right]^2 \\ &= 147 \text{ lb/ft}^2 = 1.02 \text{ lb/in}^2 \end{aligned}$$

Since this was considered to be a reasonable value for friction pressure drop, it was selected as the value to be used in designing the combustion chamber.

As previously stated, it was assumed that .25 lb air/sec had been used for cooling and carbon prevention. It was decided that the air to cool the dome should be introduced through a ring of eight 3/8-inch holes around the fuel nozzle. The air jets through these holes impinge on slanted vanes which give them a swirling action and direct them onto the dome. The pressure drop equation was used to determine whether the eight 3/8-inch holes would give the required air flow.

$$Q = \frac{A_2 C_o}{\sqrt{1 - \left(\frac{A_2}{A_1} \right)^2}} \sqrt{2g \left(\frac{\Delta P}{\rho} \right)}$$

where Q = air flow through the eight 3/8-inch holes in ft³/sec

A_2 = area of the eight 3/8-inch holes in ft²

A_1 = area of combustion chamber in ft²

$$A_2 = 8 \times \frac{\pi}{4} \left(\frac{3/8}{12} \right)^2 = .00615 \text{ ft}^2 = .885 \text{ in}^2$$

$$\left[\left(\frac{1}{\sqrt{2}} \right) \right] \frac{1}{\sqrt{2}} = \frac{1}{2} \Delta$$

DATE: 11/11/1977

$$\left[\left(\frac{100}{1000} \right)^2 \sqrt{\frac{1000}{1000000}} \right] \frac{1000000}{1000000} = \Delta$$

There is a possibility that the information obtained from the above sources may be of value in the investigation of the case.

$$\sqrt{\frac{\Delta}{g}} = \frac{\Delta}{g} = \frac{1}{g} \sqrt{\frac{\Delta}{g}}$$

There is a lot of talk about the effect of the law on the

17-00000

$$\left(\frac{1}{10} \right) \pi$$

$$A_1 = \frac{\pi}{4} \frac{7.33^2}{12} = .29 \text{ ft}^2 = 41.8 \text{ in}^2$$

$$q = \frac{.00615 \times .61}{\sqrt{1 - \left(\frac{.00615^2}{.29}\right)}} \sqrt{64.4 \left(\frac{147}{.0965}\right)}$$

$$= 1.17 \text{ ft}^3/\text{sec} \text{ or } .11 \text{ lb/sec}$$

This value seems reasonable since a portion of the .25 lb air/sec must enter through an annular slot located between the nose hemisphere and the walls of the basket.

The desired flow through this slot is then:

$$.25 - .11 = .14 \text{ lb/sec or } 1.45 \text{ ft}^3/\text{sec}$$

The pressure drop equation was used to find the slot area required to obtain this flow. Since the term $\sqrt{1 - (A_2/A_1)^2}$ will be very nearly equal to 1, the pressure drop equation may be used to solve for A_2 thus:

$$A_2 = \frac{q}{C_o \sqrt{2g(\Delta P/\rho)}}$$

where A_2 = slot area in ft^2

q = air flow through the slot in ft^3/sec

$$A_2 = \frac{1.45}{.61 \sqrt{64.4 \left(\frac{147}{.0965}\right)}}$$

$$= .0076 \text{ ft}^2 = 1.09 \text{ in}^2$$

The required thickness of the slot is found as follows:

$$A_2 = \frac{\pi}{4} [D^2 - (D - x)^2]$$

$$A_1 = \frac{\pi}{4} \left(\frac{1.0}{1.0} \right)^2 = 0.785 \text{ in}^2$$

$$\left(\frac{1.0}{1.0} \right)^2 = \left(\frac{1.0}{1.0} \right)^2$$

$$A_1 = 0.785 \text{ in}^2$$

This section is a rectangular plate of steel.

It is subject to the following stresses and strains.

Between the two supports and the ends of the plate.

The stresses are as follows:

$$A_1 = 0.785 \text{ in}^2$$

The stresses are as follows:

even though the plate is thin.

The stresses are as follows:

even though the plate is thin.

$$\left(\frac{1.0}{1.0} \right)^2 = \left(\frac{1.0}{1.0} \right)^2$$

where A_1 is the area of the plate.

The stresses are as follows:

$$\left(\frac{1.0}{1.0} \right)^2 = \left(\frac{1.0}{1.0} \right)^2$$

$$A_1 = 0.785 \text{ in}^2$$

The stresses are as follows:

where:

$$\left[\frac{\pi}{4} \left(\frac{1.0}{1.0} \right)^2 \right]$$

where D = diameter of burner basket in inches

x = clearance in inches

$$1.09 = \frac{\pi}{4} [36 - (6 - x)^2]$$

$$x = .116 \text{ in}$$

The slot thickness will then be:

$$\frac{x}{2} = \frac{.116}{2} = .058 \text{ in} \approx 1/16 \text{ in}$$

In order to get the required air flow for the primary section, three rings of eight 1/8-inch holes were required in the burner basket in addition to the ring of eight 3/8-inch holes in the dome and the annular slot.

Design of Secondary Zone

The weight flow through the primary zone is .85 lb/sec. This leaves 2.15 lb/sec to flow through the secondary zone which includes cooling louvres, the tail cone annulus, and the secondary holes.

Louvres

Air introduced through the louvres is intended to cool the basket and to prevent the formation of carbon. The louvres are shaped to deflect the air along the inner surface of the liner, thus keeping the velocity high enough to sweep away any carbon which might form. Three rings of eight louvres each were considered adequate.

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$$[F_{\alpha\beta} - A] = 0 \quad \text{if } \alpha \neq \beta$$

about 25-30 years in the past and the number of
repaired in the same period is estimated to the end of
primary section, about 1000 at about 10-15 years ago
in order to get the number of the 100

any more when someone asked how, he said that
 12/20/68. This letter was given to the woman
 The woman told through the letter was in 1968

1. The first of these is the fact that the Commission has not yet received any information from the Government of the United States regarding the activities of the Committee for the Liberation of the People of the South (CLPS) in the United States.

Each louvre is 1/2 inch long and 1/16 inch deep.

This gives an area of:

$$A = \frac{1}{16} \times \frac{1}{2} = .0312 \text{ in}^2$$

The area of one ring of eight louvres is:

$$A = .0312 \times 8 = .25 \text{ in}^2$$

Now, assuming an orifice coefficient of .61, the flow through the louvres is as follows:

$$Q = A_2 C_o \sqrt{2g \left(\frac{\Delta P}{\rho} \right)}$$

(The term $1/\sqrt{1 - (A_2/A_1)^2}$ has been omitted since in all cases it is very nearly equal to 1.)

$$\text{where } Q = \frac{.25}{144} \times .61 \sqrt{64.4 \left(\frac{147}{.0965} \right)}$$

$$= .332 \text{ ft}^3/\text{sec or } .032 \text{ lb/sec}$$

For three rings, the flow will be:

$$3 \times .032 = .096 \text{ lb/sec}$$

Tail Cone Annulus

The flow through the tail cone annulus is necessary to cool the inner liner which is in direct contact with the hot gases. The film of air need not be very large to accomplish this. The 1/16-inch clearance allowed is in accordance with current practice.

According to Reference 7, the pressure drop across the tail cone annulus is given by the following formula:

Each tower is 1/2 inch long and 1/2 inch deep.

This gives an area of:

$$A = \frac{1}{2} \times \frac{1}{2} = \frac{1}{4} \text{ sq. in.}$$

The area of one ring of silver is:

$$A = \pi R^2 - \pi r^2 = \pi (R^2 - r^2)$$

Now, assuming an average coefficient of 0.1, the area

through the silver is as follows:

$$A = \pi \left(\frac{1}{2} \right)^2 - \pi \left(\frac{1}{4} \right)^2 = \frac{3\pi}{16}$$

(The area $A = \pi \left(\frac{1}{2} \right)^2$ has been omitted since it is

smaller than the area of the silver ring.)

$$A = \frac{3\pi}{16} = \frac{3 \times 3.1416}{16} = \frac{9.4248}{16} = 0.5891 \text{ sq. in.}$$

2. Area of the silver ring.

The area of the silver ring is:

$$A = \pi R^2 - \pi r^2 = \pi (R^2 - r^2)$$

Let the area be:

The first through the silver is as follows:
to find the area of the silver ring is to find the area of the
the ring. The area of the ring is the area of the outer circle
minus the area of the inner circle. The area of the outer circle is
the area of the inner circle. The area of the inner circle is the
area of the outer circle. The area of the outer circle is the
area of the inner circle.

assuming the area of the silver ring is the area of the outer circle
minus the area of the inner circle. The area of the outer circle is
the area of the inner circle. The area of the inner circle is the
area of the outer circle. The area of the outer circle is the
area of the inner circle.

$$\frac{\Delta P}{\rho} = \frac{f}{4} \frac{h}{R} \frac{v^2}{2g}$$

where ΔP = friction pressure drop in lb/ft² =
147 lb/ft²

ρ = density of air in lb/ft³ = .0965 lb/ft³

f = friction factor

h = length of tail cone in ft = 5/12 ft

R = hydraulic radius in ft

v = average velocity through annular space
in ft/sec

Reference 3 gives the relation between f and Reynold's number, NR . NR , in turn, is dependent upon v . Thus, it can be seen that a cut and try calculation is necessary to arrive at values of f and v which will satisfy the above equation. From Reference 7,

$$NR = \frac{4vR}{\mu}$$

where μ = viscosity in slugs/ft-sec

$$R = \frac{A_{ave}}{W}$$

Where A_{ave} = average cross-sectional area in ft²

W = wetted perimeter in ft

$$A_{ave} = .785 \left[\frac{[(6.125)^2 - 6^2] + [(4.125)^2 - 4^2]}{2} \right]$$

$$= 1.023 \text{ in}^2 = .00711 \text{ ft}^2$$

$$\Delta = \frac{1}{2} \left(\frac{1}{\mu} + \frac{1}{\nu} \right)$$

where Δ = friction coefficient due to rolling =

$$\frac{1}{2} \left(\frac{1}{\mu} + \frac{1}{\nu} \right)$$

ρ = density of air in lb/cu ft = 0.0012

L = friction factor =

μ = coefficient of friction in lb/cu ft

ν = coefficient of friction in lb/cu ft

μ = coefficient of friction through member space

in lb/cu ft

where μ = friction factor between μ and

Reynolds's number, Re . μ is given by equation 10.1

Then, it can be seen that μ and ν coefficient are not

always as given in Table 10.1 and μ will vary

the above equation. Thus, μ is given by

$$\mu = \frac{1}{2} \left(\frac{1}{\mu} + \frac{1}{\nu} \right)$$

where μ is given by equation 10.1

$$\mu = \frac{1}{2} \left(\frac{1}{\mu} + \frac{1}{\nu} \right)$$

where μ is given by equation 10.1

where μ is given by equation 10.1

$$\left[\frac{1}{2} \left(\frac{1}{\mu} + \frac{1}{\nu} \right) \right] = \frac{1}{2} \left(\frac{1}{\mu} + \frac{1}{\nu} \right)$$

where μ is given by equation 10.1

$$EP = \pi (D_2 + D_1)$$

where D_1 = inside diameter of section at A_{ave} in ft

D_2 = outside diameter of section at A_{ave} in ft

$$D_2 = D_1 + \frac{.125}{12}$$

$$.00711 = .785 \left[\left(D_1 + \frac{.125}{12} \right)^2 - D_1^2 \right]$$

$$D_1 = .43 \text{ ft}$$

$$D_2 = .44 \text{ ft}$$

$$EP = \pi (.43 + .44) = 2.735 \text{ ft}$$

$$R = \frac{.00711}{2.735} = .0026 \text{ ft}$$

Assuming $v = 250 \text{ ft/sec}$:

$$\begin{aligned} NR &= \frac{4 \times 250 \times .0965 \times .0026}{.45 \times 10^{-6} \times 32.2} \\ &= 17320 \end{aligned}$$

$$\text{and } f = .042$$

Solving for v to check assumption:

$$\frac{147}{.0965} = \frac{.042 \left(\frac{5/12}{4} \right) v^2}{.0026 \times 64.4}$$

$$v = 242 \text{ ft/sec}$$

Assuming $v = 240 \text{ ft/sec}$ and recalculating:

$$NR = 17320 \times \frac{240}{250} = 16660$$

$$\text{and } f = .0435$$

$$(1^2 + 2^2) \pi = 5\pi$$

Let $\alpha_1, \alpha_2, \dots, \alpha_n$ be the roots of the equation $x^n - 1 = 0$.

Then $\alpha_1, \alpha_2, \dots, \alpha_n$ are the n th roots of unity.

$$\alpha_1 + \alpha_2 + \dots + \alpha_n = 0$$

$$\left[\frac{1}{2} \alpha_1 + \frac{1}{2} \alpha_2 + \dots + \frac{1}{2} \alpha_n \right] \cos \theta + i \left[\frac{1}{2} \alpha_1 + \frac{1}{2} \alpha_2 + \dots + \frac{1}{2} \alpha_n \right] \sin \theta = 0$$

$$\cos \theta = 0$$

$$\sin \theta = 0$$

$$\cos \theta = 0 \Rightarrow \theta = \frac{\pi}{2}, \frac{3\pi}{2}, \dots$$

$$\sin \theta = 0 \Rightarrow \theta = 0, \pi, 2\pi, \dots$$

$$\cos \theta = 0 \Rightarrow \theta = \frac{\pi}{2}, \frac{3\pi}{2}, \dots$$

$$\sin \theta = 0 \Rightarrow \theta = 0, \pi, 2\pi, \dots$$

$$\cos \theta = 0 \Rightarrow \theta = \frac{\pi}{2}, \frac{3\pi}{2}, \dots$$

$$\sin \theta = 0$$

$$\cos \theta = 0$$

$$\sin \theta = 0 \Rightarrow \theta = 0, \pi, 2\pi, \dots$$

$$\frac{1}{2} \alpha_1 + \frac{1}{2} \alpha_2 + \dots + \frac{1}{2} \alpha_n = 0$$

$$\cos \theta = 0$$

$$\sin \theta = 0 \Rightarrow \theta = 0, \pi, 2\pi, \dots$$

$$\cos \theta = 0 \Rightarrow \theta = \frac{\pi}{2}, \frac{3\pi}{2}, \dots$$

$$\sin \theta = 0$$

$$v = 242 \sqrt{\frac{.042}{.0435}} = 238 \text{ ft/sec}$$

This checks the assumed value. Then,

$$Q = A_{ave} v$$

where Q = flow through tail cone annulus in ft^3/sec

$$Q = .00711 \times 240$$

$$= 1.707 \text{ ft}^3/\text{sec} \text{ or } .165 \text{ lb/sec}$$

Secondary Holes

Knowing the flow through the primary zone, the louvres, and the tail cone annulus, the weight flow through the secondary holes can be obtained.

$$\begin{array}{r} .85 \text{ lb/sec primary air} \\ .165 \text{ lb/sec tail cone annulus} \\ \underline{.096 \text{ lb/sec louvres}} \\ 1.111 \text{ lb/sec total} \end{array}$$

Then, the weight flow through the secondary holes is

$$3.000 - 1.111 = 1.889 \text{ lb/sec}$$

Because of the high velocities in the combustion chamber, it was assumed that the air would flow through the secondary holes at some angle other than 90° . This angle was taken to be 45° throughout the secondary zone. Appendix A presents an experiment which was the basis for this assumption.

This angularity of flow necessitates the introduction of an effectiveness coefficient, C_e , in addition to the orifice coefficient, C_o , when calculating flow through

a secondary hole. C_e is defined as the sine of the angle of flow.

$$C_e = \sin 45^\circ$$

$$= .707$$

To find the flow through a ring of eight 5/8-inch secondary holes:

$$Q = N_h A_2 C_e C_o \sqrt{2g \left(\frac{\Delta P}{\rho} \right)}$$

where Q = air flow through ring of holes in ft^3/sec

N_h = number of holes in ring = 8

A_2 = area of each hole = .00213 ft^2

C_e = effectiveness coefficient = .707

C_o = orifice coefficient = .61

g = gravitational constant = 32.2 ft/sec^2

ΔP = friction pressure drop = 147 lb/ft^2

ρ = density of air = .0965 lb/ft^3

$$Q = 8 \times .00213 \times .707 \times .61 \sqrt{64.4 \left(\frac{147}{.0965} \right)}$$

$$= 2.29 \text{ ft}^3/\text{sec} \text{ or } .221 \text{ lb/sec}$$

Similarly, for a ring of eight 11/16-inch holes is 2.88 ft^3/sec or .278 lb/sec . Thus, six rings of 11/16-inch holes and one ring of 5/8-inch holes will admit the 1.859 lb/sec of secondary air that remains.

* courtesy of the author

of 1.5

PLATE 3

The first two large channels in the right side of the

$$\left(\frac{\Delta}{\rho}\right)^{1/2}$$

There is a lot of talk about the importance of the

[illegible]

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DATE: 08-29-2007

It is a pleasure to have you here, and we hope you will find the trip well worth the effort.

ALL INFORMATION CONTAINED HEREIN IS UNCLASSIFIED

1. May 1944 - 2. May 1944

Check of Air Flows

	.11	lb/sec holes in hemisphere	
	.14	" annular slot	
3 x .2 =	.60	" three rings of 1/2-inch holes	
6 x .278 =	1.67	" six rings of 11/16-inch holes	
	.221	" one ring of 5/8-inch holes	
	.165	" tail cone annulus	
	.096	" louvres	

3.002 lb/sec total air flow	<u>check</u>
-----------------------------	--------------

Location of Holes

The primary air should be introduced far enough downstream to insure the development of a strong combustion eddy. This is about three-fourths to one diameter from the fuel nozzle.

The secondary air should be first introduced at a low rate and then at an increasingly greater rate downstream. This is to prevent chilling of the flame front and the resulting stoppage of the combustion process. By introducing the first part of the secondary air six to eight inches from the fuel nozzle, enough time is allowed for the combustion to be well along towards completion.

A convenient method for locating the primary and secondary holes to satisfy these requirements is shown in Figure 6 in which total area of the openings into the basket

Table 1.1.1

1.1.1.1	1.1.1.1.1	1.1.1.1.1.1	1.1.1.1.1.1.1
1.1.1.2	1.1.1.2.1	1.1.1.2.1.1	1.1.1.2.1.1.1
1.1.1.3	1.1.1.3.1	1.1.1.3.1.1	1.1.1.3.1.1.1
1.1.1.4	1.1.1.4.1	1.1.1.4.1.1	1.1.1.4.1.1.1
1.1.1.5	1.1.1.5.1	1.1.1.5.1.1	1.1.1.5.1.1.1
1.1.1.6	1.1.1.6.1	1.1.1.6.1.1	1.1.1.6.1.1.1
1.1.1.7	1.1.1.7.1	1.1.1.7.1.1	1.1.1.7.1.1.1
1.1.1.8	1.1.1.8.1	1.1.1.8.1.1	1.1.1.8.1.1.1
1.1.1.9	1.1.1.9.1	1.1.1.9.1.1	1.1.1.9.1.1.1
1.1.1.10	1.1.1.10.1	1.1.1.10.1.1	1.1.1.10.1.1.1

1.1.1.11 1.1.1.11.1 1.1.1.11.1.1 1.1.1.11.1.1.1

Table 1.1.2

The following table shows the results of the experiments conducted in the laboratory of the Department of Physics, University of Toronto, during the year 1928-29. The results are given in the form of a table, and the units of measurement are given in the footnotes.

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TOTAL ENTRANCE AREA INTO BURNING BASKET - SQ. IN.

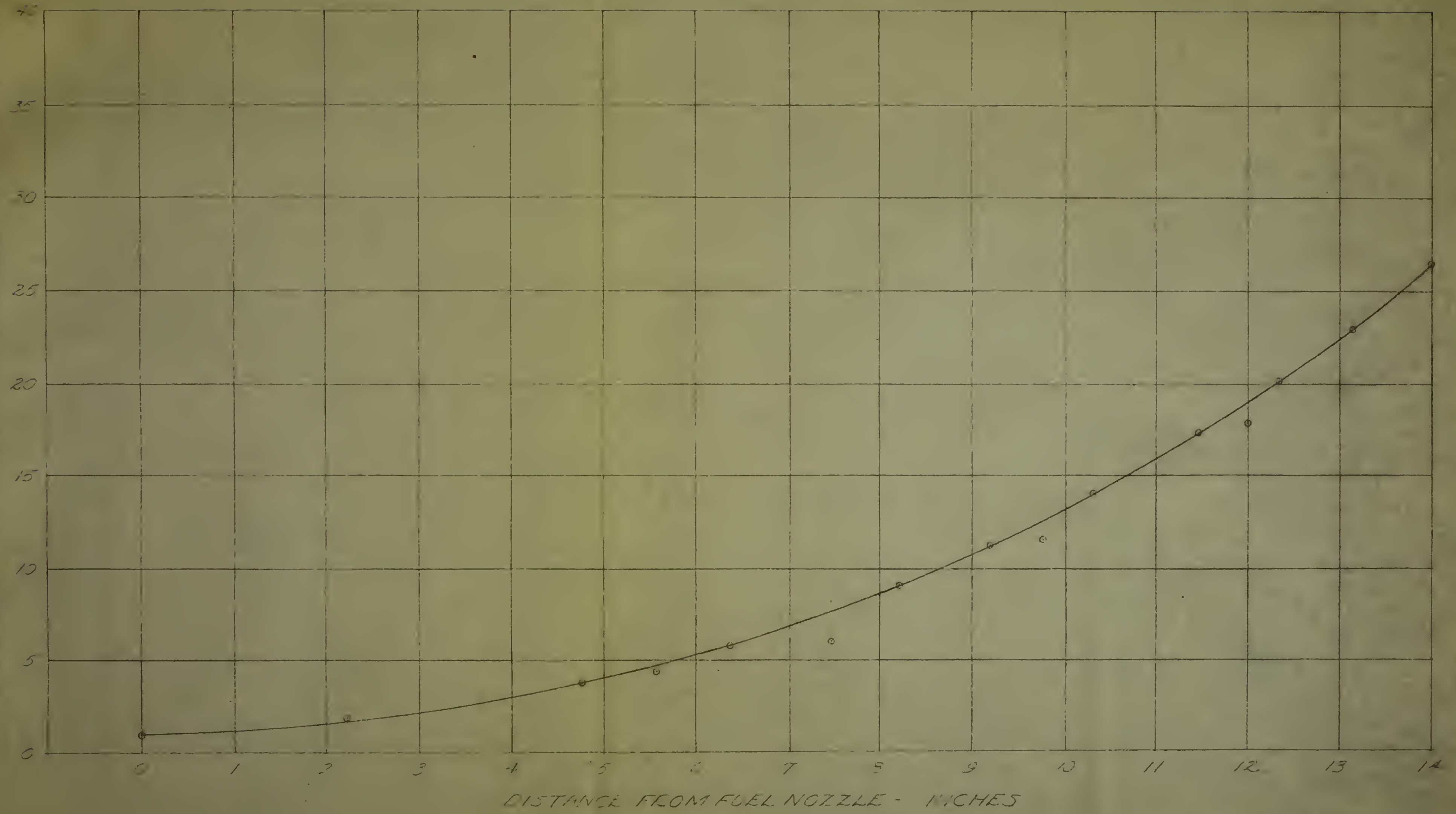


FIGURE 6
CURVE USED IN LOCATING PRIMARY AND SECONDARY HOLES.

is plotted vs. distance from the fuel nozzle. The curve shown in an arc of a circle and is defined as follows:

1. The center of the circle lies on the vertical line through the zero distance point.
2. One point on the arc is the area of the openings into the burner basket ahead of the fuel nozzle, i.e., the area of the eight $3/8$ -inch holes. This area is plotted at the point of zero distance.
3. The other point of the arc is the total area of all the openings, slots, and louvres in the burner basket. This area is plotted at the distance at which the last row of secondary holes is desired.

Now, by taking the areas of the rings of holes and fitting them to this curve, a satisfactory distribution will automatically follow.

This method, used judiciously, should take much of the cut and try out of burner design.

The cooling louvres can now be placed in spots where the boundary layer would otherwise thicken up and let carbon deposits form.

Figure 7 shows the final configuration of the combustion chamber.

is placed in position from the first position. The curve shown in the eye at a distance of 100 mm is shown in Figure 1.

1. The curve of the curve line in the vertical line through the eye is shown in Figure 1.

2. The curve of the curve line in the horizontal line through the eye is shown in Figure 2. This curve is placed at the point of zero distance.

3. The curve of the curve line in the vertical line through the eye is shown in Figure 3. This curve is placed at the point of zero distance.

4. The curve of the curve line in the horizontal line through the eye is shown in Figure 4. This curve is placed at the point of zero distance.

5. The curve of the curve line in the vertical line through the eye is shown in Figure 5. This curve is placed at the point of zero distance.

6. The curve of the curve line in the horizontal line through the eye is shown in Figure 6. This curve is placed at the point of zero distance.

7. The curve of the curve line in the vertical line through the eye is shown in Figure 7. This curve is placed at the point of zero distance.

8. The curve of the curve line in the horizontal line through the eye is shown in Figure 8. This curve is placed at the point of zero distance.

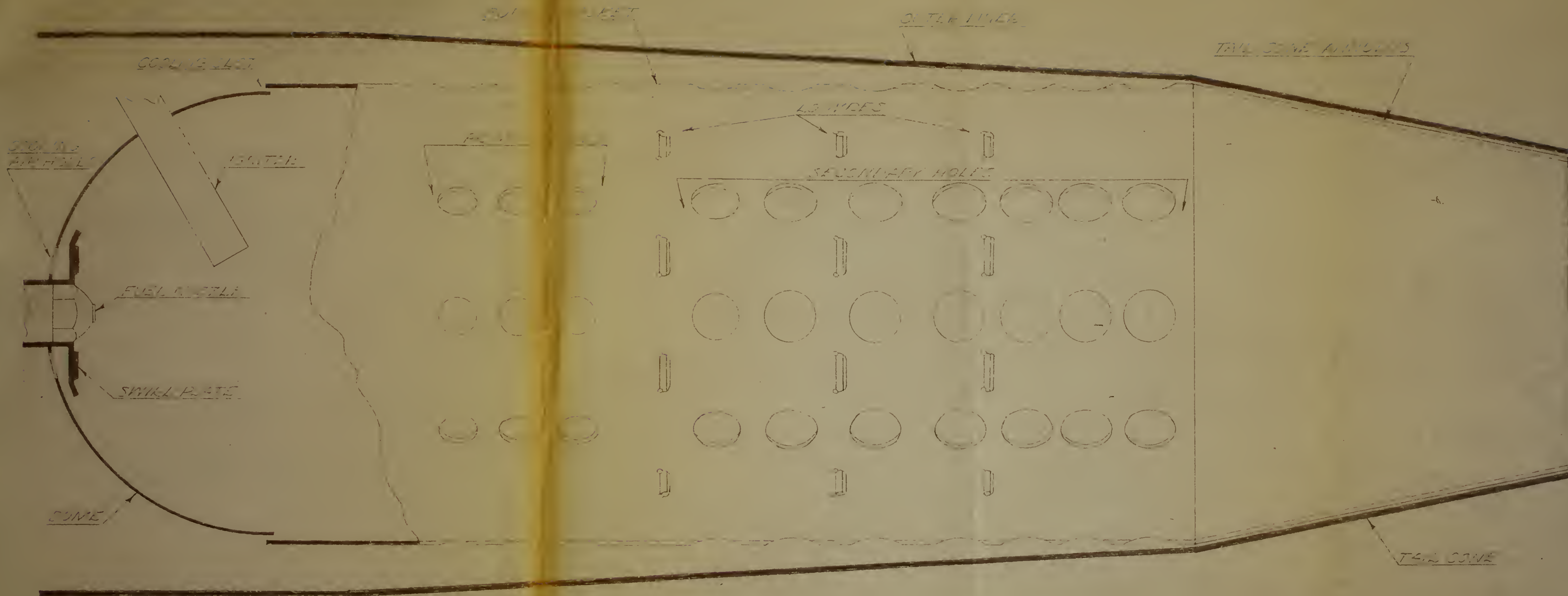


FIGURE 7 DESIGNED COMBUSTION CHAMBER.

FULL SCALE

VII. CALCULATION OF PERFORMANCE

Friction Pressure Drop

Since the flow through the combustion chamber is parallel, the pressure drop through any one part is equal to the pressure drop through the entire burner. The friction pressure drop, as calculated in the preceding section, is:

$$\Delta P = 147 \text{ lb/ft}^2 \text{ or } 1.02 \text{ lb/in}^2$$

Momentum Pressure Drop

Having the friction pressure loss in hand, it is now necessary to determine the pressure loss due to combustion in order to have the total pressure loss through the burner.

The assumption is now made that the combustion takes place in a burner of constant cross-sectional area and that no mass is added by the fuel.

In a combustion process, the impulse-momentum law holds. This law states that "impulse is equal to the change in momentum." Written mathematically, this is:

$$Ft = \Delta (Mv)$$

where F = pounds force

t = time in sec

M = pounds mass

v = velocity in ft/sec

VII. CALCULATION OF EQUATIONS

Electric Circuit Law

Using the law of conservation of energy, the electrical energy is converted into mechanical energy and vice versa. The total energy of the system is constant. The electrical energy is converted into mechanical energy and vice versa. The total energy of the system is constant.

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$$\Delta \epsilon = \epsilon_1 - \epsilon_2 = \epsilon_1 - \epsilon_2$$

Mechanical Circuit Law

Using the law of conservation of energy, the mechanical energy is converted into electrical energy and vice versa. The total energy of the system is constant. The mechanical energy is converted into electrical energy and vice versa. The total energy of the system is constant.

In a mechanical system, the energy is converted into electrical energy and vice versa. The total energy of the system is constant. The mechanical energy is converted into electrical energy and vice versa. The total energy of the system is constant.

$$\Delta \epsilon = \epsilon_1 - \epsilon_2 = \epsilon_1 - \epsilon_2$$

where ϵ = electrical energy

ϵ = electrical energy

ϵ = electrical energy

ϵ = electrical energy

Now, since mass is assumed constant:

$$Ft = M\Delta v$$

$$F = \frac{M\Delta v}{t}$$

$$F = \frac{w}{g}(v_5 - v_4)$$

where w = weight flow in lb/sec

g = ratio of absolute to gravitational units
of mass

Pressure is force per unit area, and since the burner is of constant area,

$$AP_{4s} - AP_{5s} = \frac{w}{g}(v_5 - v_4)$$

where P_{4s} = entering static pressure in lb/ft²

P_{5s} = exit static pressure in lb/ft²

A = burner area in ft²

In the burner at hand, the known conditions are: entering total pressure, P_{4t} ; entering velocity, v_4 ; entering total temperature, T_{4t} ; and exit total temperature, T_{5t} . In order to obtain pressure drop due to combustion, it is necessary to know the theoretical exit total pressure, P_{5t} .

From adiabatic considerations, it is known that:

$$\frac{P_{4t}}{P_{4s}} = \left(\frac{T_{4t}}{T_{4s}} \right)^{\frac{\gamma}{\gamma-1}}$$

$$\text{and } \frac{P_{5t}}{P_{5s}} = \left(\frac{T_{5t}}{T_{5s}} \right)^{\frac{\gamma}{\gamma-1}}$$

(Note: when using the above formulae)

$$\Delta t = \Delta t_0 \sqrt{1 - \frac{v^2}{c^2}}$$

$$\Delta x = \Delta x_0 \sqrt{1 - \frac{v^2}{c^2}}$$

$$\left(\frac{\Delta x}{\Delta t} \right)^2 = \left(\frac{\Delta x_0}{\Delta t_0} \right)^2 \left(1 - \frac{v^2}{c^2} \right)$$

where v = velocity of light in vacuum

Δx = distance between two points in space

of time

Proportion is large for small v , and since the error

is of constant order

$$\Delta t = \Delta t_0 \left(1 - \frac{v^2}{c^2} \right)^{1/2}$$

where Δt_0 = proper time interval in Δx_0

Δx_0 = distance between two points in space

and Δt_0 = proper time interval in Δx_0

In the present case, the proper distance and the

proper time interval are both constant, and the

total distance and total time are both constant, and

the proper distance and proper time are both constant, and

the proper distance and proper time are both constant,

and

the proper distance and proper time are both constant, and

$$\left(\frac{\Delta x}{\Delta t} \right)^2 = \left(\frac{\Delta x_0}{\Delta t_0} \right)^2 \left(1 - \frac{v^2}{c^2} \right)$$

$$\left(\frac{\Delta x}{\Delta t} \right)^2 = \left(\frac{\Delta x_0}{\Delta t_0} \right)^2 \left(1 - \frac{v^2}{c^2} \right)$$

Then,
$$\frac{P_{5t}}{P_{4t}} = \frac{P_{5s}}{P_{4s}} \left(\frac{T_{5t}}{T_{4t}} \right)^{\frac{\gamma}{\gamma-1}} \left(\frac{T_{4s}}{T_{5s}} \right)^{\frac{\gamma}{\gamma-1}}$$

Since
$$P/\rho = RT$$

where P = pressure in lb/ft²

ρ = density in lb/ft³

R = gas constant in ft²/°R

T = temperature in °R

and

$$Q = w/\rho$$

where Q = volume flow in ft³/sec

then

$$\frac{PQ}{w} = RT$$

and

$$\frac{P_{4s} Q_4}{T_{4s}} = R w = \frac{P_{5s} Q_5}{T_{5s}}$$

Also,

$$\frac{P_{4s} v_4}{T_{4s}} = \frac{P_{5s} v_5}{T_{5s}}$$

So,
$$\frac{P_{5t}}{P_{4t}} = \frac{v_4}{v_5} \left(\frac{T_{5t}}{T_{4t}} \right)^{\frac{\gamma}{\gamma-1}} \left(\frac{T_{4s}}{T_{5s}} \right)^{\frac{\gamma}{\gamma-1}}$$

To find static temperatures:

$$\frac{T_{4t}}{T_{4s}} = 1 + \frac{\gamma-1}{2} M_4^2$$

and
$$M_4^2 = \frac{v_4^2}{\gamma R T_{4s}}$$

$$\frac{1}{2} \left(\frac{1}{2} \right)^{\frac{1}{2}} \left(\frac{1}{2} \right)^{\frac{1}{2}} = \frac{1}{2}$$

$$\frac{1}{2} = \frac{1}{2}$$

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$$\frac{1}{2} = \frac{1}{2}$$

$$\frac{1}{2} = \frac{1}{2}$$

$$\frac{T_{4t}}{T_{4s}} = 1 + \frac{\gamma - 1}{2} \left(\frac{v_4^2}{\gamma R T_{4s}} \right)$$

$$T_{4t} = T_{4s} + \frac{\gamma - 1}{2} \left(\frac{v_4^2}{\gamma R} \right)$$

$$\text{Then, } T_{4s} = T_{4t} - \frac{v_4^2}{2\gamma R \left(\frac{\gamma}{\gamma - 1} \right)}$$

Similarly, T_{5s} is obtained.

To find v_5 :

$$AP_{4s} - AP_{5s} = \frac{\gamma}{\gamma - 1} (v_5 - v_4)$$

$$A = Q/v$$

$$\frac{P_{4s} Q_4}{v_4} - \frac{P_{5s} Q_5}{v_5} = \frac{\gamma}{\gamma - 1} (v_5 - v_4)$$

$$P/\rho = RT$$

$$Q = w/\rho$$

$$PQ = RTw$$

$$\text{Now, } \frac{RT_{4s} w}{v_4} - \frac{RT_{5s} w}{v_5} = \frac{\gamma}{\gamma - 1} (v_5 - v_4)$$

$$\frac{T_{4s}}{v_4} - \frac{T_{5s}}{v_5} = \frac{1}{R\gamma} (v_5 - v_4)$$

$$\frac{T_{4s}}{v_4} = \frac{T_{4t}}{v_4} - \frac{v_4}{2gR \left(\frac{\gamma}{\gamma-1} \right)}$$

$$\frac{T_{5s}}{v_5} = \frac{T_{5t}}{v_5} - \frac{v_5}{2gR \left(\frac{\gamma}{\gamma-1} \right)}$$

$$\frac{T_{4s}}{v_4} - \frac{T_{5s}}{v_5} = \frac{T_{4t}}{v_4} - \frac{T_{5t}}{v_5} + \frac{v_5 - v_4}{2gR \left(\frac{\gamma}{\gamma-1} \right)}$$

$$\frac{T_{4t}}{v_4} - \frac{T_{5t}}{v_5} + \frac{v_5 - v_4}{2gR \left(\frac{\gamma}{\gamma-1} \right)} = \frac{1}{Rg} (v_5 - v_4)$$

$$\frac{T_{4t}}{v_4} - \frac{T_{5t}}{v_5} = \frac{1}{Rg} (v_5 - v_4) - \frac{v_5 - v_4}{2gR \left(\frac{\gamma}{\gamma-1} \right)}$$

$$= (v_5 - v_4) \left[\frac{1}{Rg} - \frac{1}{2gR \left(\frac{\gamma}{\gamma-1} \right)} \right]$$

$$= (v_5 - v_4) \left[\frac{2\gamma - (\gamma-1)}{2\gamma Rg} \right]$$

$$= (v_5 - v_4) \left(\frac{\gamma+1}{2\gamma Rg} \right)$$

$$\frac{T_{4t}}{v_4} \left[1 - \frac{T_{5t}/v_5}{T_{4t}/v_4} \right] = \left(\frac{\gamma+1}{2\gamma Rg} \right) v_4 \left(\frac{v_5}{v_4} - 1 \right)$$

$$\frac{T_{4t}}{v_4} \left[1 - \frac{T_{5t}}{T_{4t}} \frac{v_4}{v_5} \right] = \left(\frac{\gamma+1}{2\gamma Rg} \right) v_4 \left(\frac{v_5}{v_4} - 1 \right)$$

$$\left(\frac{1}{1-\gamma}\right)^{\frac{1}{1-\gamma}} = \frac{1}{1-\gamma} = \frac{1}{1-\gamma}$$

$$\left(\frac{1}{1-\gamma}\right)^{\frac{1}{1-\gamma}} = \frac{1}{1-\gamma} = \frac{1}{1-\gamma}$$

$$\left(\frac{1}{1-\gamma}\right)^{\frac{1}{1-\gamma}} + \frac{1}{1-\gamma} = \frac{1}{1-\gamma} + \frac{1}{1-\gamma} = \frac{1}{1-\gamma}$$

$$(1-\gamma)^{\frac{1}{1-\gamma}} = \left(\frac{1}{1-\gamma}\right)^{\frac{1}{1-\gamma}} = \frac{1}{1-\gamma} = \frac{1}{1-\gamma}$$

$$\left(\frac{1}{1-\gamma}\right)^{\frac{1}{1-\gamma}} = (1-\gamma)^{\frac{1}{1-\gamma}} = \frac{1}{1-\gamma} = \frac{1}{1-\gamma}$$

$$\left[\left(\frac{1}{1-\gamma}\right)^{\frac{1}{1-\gamma}} = \frac{1}{1-\gamma}\right] (1-\gamma) = 1$$

$$\left[\frac{1-\gamma-\gamma}{1-\gamma}\right] (1-\gamma) = 1$$

$$\left(\frac{1-\gamma}{1-\gamma}\right) (1-\gamma) = 1$$

$$(1-\gamma)^{\frac{1}{1-\gamma}} \left(\frac{1-\gamma}{1-\gamma}\right) = \left[\frac{1-\gamma}{1-\gamma} = 1\right] \frac{1-\gamma}{1-\gamma}$$

$$(1-\gamma)^{\frac{1}{1-\gamma}} \left(\frac{1-\gamma}{1-\gamma}\right) = \left[\frac{1-\gamma}{1-\gamma} = 1\right] \frac{1-\gamma}{1-\gamma}$$

$$1 - \frac{T_{5t}}{T_{4t}} \frac{v_4}{v_5} = \left(\frac{\gamma+1}{2} \right) \left(\frac{v_4^2}{\gamma R T_{4t}} \right) \left(\frac{v_5}{v_4} - 1 \right)$$

$$= \left(\frac{\gamma+1}{2} \right) M_{4t}^2 \left(\frac{v_5}{v_4} - 1 \right)$$

$$= \left(\frac{\gamma+1}{2} \right) M_{4t}^2 \left(\frac{v_5}{v_4} \right) - \left(\frac{\gamma+1}{2} \right) M_{4t}^2$$

$$1 + \left(\frac{\gamma+1}{2} \right) M_{4t}^2 = \left(\frac{\gamma+1}{2} \right) M_{4t}^2 \left(\frac{v_5}{v_4} \right) + \frac{T_{5t}/T_{4t}}{v_5/v_4}$$

$$(v_5/v_4)^2 \left(\frac{\gamma+1}{2} \right) M_{4t}^2 - \frac{v_5}{v_4} \left[1 + \left(\frac{\gamma+1}{2} \right) M_{4t}^2 \right] + \frac{T_{5t}}{T_{4t}} = 0$$

let $a = \frac{\gamma+1}{2} M_{4t}^2$ and $b = \frac{T_{5t}}{T_{4t}}$

$$a \left(\frac{v_5}{v_4} \right)^2 - (1+a) \frac{v_5}{v_4} + b = 0$$

$$\frac{v_5}{v_4} = \frac{1+a \pm \sqrt{(1+a)^2 - 4ab}}{2a}$$

then,

$$\frac{v_5}{v_4} = \frac{1 + \frac{\gamma+1}{2} M_{4t}^2 \pm \sqrt{1 + \frac{\gamma+1}{2} M_{4t}^2 - 4 \left(\frac{\gamma+1}{2} M_{4t}^2 \right) \frac{T_{5t}}{T_{4t}}}}{(\gamma+1) M_{4t}^2}$$

$$\left(1 - \frac{\delta}{\lambda^2}\right)^{2\lambda^2} \left(\frac{\lambda + \delta}{\lambda}\right) =$$

$$f\left(\frac{x}{a}\right) = \left(\frac{a}{x}\right)^2 f\left(\frac{x}{a}\right) =$$

$$\frac{10^5 \times 10^2}{10^3 \times 10^4} = \left(\frac{10^2}{10^4}\right)^2 \times 10^2 \left(\frac{10^8}{10^4}\right) = 10^2 \left(\frac{10^8}{10^4}\right) = 1$$

$$C = \frac{dQ}{dT} = \left[16^{\frac{1}{2}} \left(\frac{1-\gamma}{2} \right) + 1 \right] \frac{dQ}{dT} = 16^{\frac{1}{2}} \left(\frac{1-\gamma}{2} \right)^{\frac{1}{2}} (4 \ln 2)^{\frac{1}{2}}$$

$$\frac{1}{2} \frac{d^2}{dt^2} = 0 \quad \text{then} \quad \frac{1}{2} \frac{d^2}{dt^2} = 0 \quad \text{and} \quad \frac{1}{2} \frac{d^2}{dt^2} = 0$$

$$C = \frac{1}{2} \left(\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} \right) = \frac{1}{2} \nabla^2$$

$$\frac{d}{dt} \left(\frac{1}{2} m v^2 \right) = \frac{d}{dt} \left(\frac{1}{2} m \left(\frac{dx}{dt} \right)^2 \right) = m \frac{dx}{dt} \frac{d^2x}{dt^2}$$

$$\frac{d^2}{dt^2} \left(\frac{\partial L}{\partial \dot{x}} \right) = \frac{\partial L}{\partial x}$$

Substituting the following values in the formula derived for

v_5 :

$$v_4 = 100 \text{ ft/sec}$$

$$M_{4t} = .0794$$

$$\gamma = 1.4$$

$$M_{4t}^2 = .0063$$

$$\gamma + 1 = 2.4$$

$$\frac{T_{5t}}{T_{4t}} = \frac{1960}{660} = 2.97$$

$$\frac{\gamma + 1}{2} = 1.2$$

we find,

$$\frac{v_5}{100} = \frac{1 + 1.2(.0063) \pm \sqrt{[1 + 1.2(.0063)]^2 - 4(1.2)(.0063)(2.97)}}{2.4(.0063)}$$

$$v_5 = 302 \text{ ft/sec}$$

Substituting the following values in the formulas derived

for T_{4s} and T_{5s} :

$$v_4 = 100 \text{ ft/sec}$$

$$v_5 = 302 \text{ ft/sec}$$

$$g = 32.2 \text{ ft/sec}^2$$

$$\gamma = 1.4$$

$$T_{4t} = 660^\circ \text{ R}$$

$$T_{5t} = 1960^\circ \text{ R}$$

$$R = 53.3 \text{ ft/}^\circ\text{R}$$

we find,

$$T_{4s} = 660 - \frac{(100)^2}{2(32.2) \left(\frac{1.4}{1.4 - 1} \right) 53.3}$$

$$= 659.2^\circ \text{ R}$$

$$\text{and } T_{5s} = 1960 - \frac{(302)^2}{2(32.2) \left(\frac{1.4}{1.4 - 1} \right) 53.3}$$

$$= 1952.4^\circ \text{ R}$$

Substituting the following values in the formula derived for P_{5t} :

$$\begin{aligned} P_{4t} &= 1.6 \text{ atm} & \gamma &= 1.4 \\ v_4 &= 100 \text{ ft/sec} & v_5 &= 302 \text{ ft/sec} \\ T_{4t} &= 660^\circ \text{ R} & T_{5t} &= 1960^\circ \text{ R} \\ T_{4s} &= 659.2^\circ \text{ R} & T_{5s} &= 1952.4^\circ \text{ R} \end{aligned}$$

we find,

$$\begin{aligned} \frac{P_{5t}}{1.6} &= \frac{\left(\frac{100}{302} \right) \left(\frac{1960}{660} \right)^{\frac{1.4}{1.4-1}}}{\left(\frac{1952.4}{659.2} \right)^{\frac{1}{1.4-1}}} \\ P_{5t} &= 1.52 \text{ atm} \end{aligned}$$

Then, the momentum pressure drop is:

$$\begin{aligned} \Delta P &= P_{4t} - P_{5t} \\ &= 1.6 - 1.52 \\ &= .08 \text{ atm} = 1.175 \text{ lb/in}^2 \end{aligned}$$

Total Pressure Drop

The total pressure drop is the sum of the friction pressure drop and the momentum pressure drop.

$$\begin{aligned} \text{Friction pressure drop} &= 1.02 \text{ lb/in}^2 \\ \text{Momentum pressure drop} &= 1.175 \text{ lb/in}^2 \\ \hline \text{Total pressure drop} &= 2.195 \text{ lb/in}^2 \end{aligned}$$

Revised the following values for the various sections:

for 1941

$1.1 = 1.1$	$1.1 = 1.1$
$1.1 = 1.1$	$1.1 = 1.1$
$1.1 = 1.1$	$1.1 = 1.1$
$1.1 = 1.1$	$1.1 = 1.1$
$1.1 = 1.1$	$1.1 = 1.1$

as follows

$$\frac{(1.1)(1.1)}{(1.1)(1.1)} = \frac{1.1}{1.1}$$

for 1.1 = 1.1

Then, the various pressure drops are

$$1.1 = 1.1 \Delta$$

$$1.1 = 1.1$$

$$1.1 = 1.1$$

total pressure drop

The total pressure drop is the sum of the various

pressure drops and the various pressure drops

pressure drops are 1.1, 1.1, 1.1

pressure drops are 1.1, 1.1, 1.1

total pressure drop is 1.1, 1.1, 1.1

For ease in comparing burners of various sizes and capacities, the parameter $\Delta P/P_{4t}$ is used. This value usually lies between four per cent and seven per cent.

$$\frac{\Delta P}{P_{4t}} = \frac{2.195}{1.6 \times 14.7} = .0934 \text{ or } 9.34\%$$

This value is a little high but is explained by the very low value of P_{4t} .

Reference 3 states that the total pressure drop should be in the neighborhood of 20 to 30 times the value of the velocity head q at point 4.

$$\begin{aligned} q &= \frac{\rho v^2}{2g} \\ &= \frac{.0965(100)^2}{64.4} \\ &= 15 \text{ lb/ft}^2 \text{ or } .104 \text{ lb/in}^2 \end{aligned}$$

$$\frac{\Delta P}{q} = \frac{2.195}{.104} = 21.1$$

Thus it is seen that the pressure drop obtained is in accord with Reference 3.

Combustion Efficiency

In an unpublished paper, Mr. E. R. Hawthorne, in collaboration with Professor H. C. Hottel of the Massachusetts Institute of Technology, presented an empir-

ical formula for combustion efficiency. This formula was derived from curves of a series of combustion chambers showing the correlation between combustion efficiency, heat input, and diameter of chamber. The formula is as follows:

$$1 - \eta = e^{\frac{-6 \times 10^6}{ID}}$$

where η = combustion efficiency

e = natural log base = 2.718

I = combustion intensity in Btu/hr-ft³-atm

D = diameter of combustion chamber in ft

$$\begin{aligned} 1 - \eta &= (2.718)^{\frac{-6 \times 10^6}{8 \times 10^6 \times .61}} \\ &= (2.718)^{-1.235} \\ &= \frac{1}{3.44} = .2905 \end{aligned}$$

$$\eta = 1 - .2905$$

$$= .7095 \text{ or } 70.95\%$$

This is a low combustion efficiency. However, this can be attributed to the fact that the very low value of P_{4t} necessitates a very high combustion intensity. The effect of combustion intensity can be shown by substituting an intensity of 4×10^6 Btu/hr-ft³-atm in the empirical formula. In this case, the efficiency jumps to 92%. In order to obtain a combustion intensity of 4×10^6 , P_{4t} must equal 3.2 atmospheres. Even this value of P_{4t} is considered quite low for modern turbo-jet engines.

and formula for calculating efficiency. This formula was
derived from a series of experiments showing that
the correlation between calculated efficiency and in-
put and diameter of member. The formula is as follows:

$$\frac{P}{A} = \frac{E}{L} \left(\frac{d}{D} \right)^4$$

where P = compression efficiency

E = modulus of elasticity = 29,000,000

L = length of member in inches = 120

d = diameter of member in inches = 1.5

$$\frac{P}{A} = \frac{29,000,000}{120} \left(\frac{1.5}{12} \right)^4$$

$$P/A = 1.125$$

$$P = 1.125 \times A$$

$$P = 1.125 \times 1.5^2$$

$$P = 2.531$$

This is a low efficiency efficiency. However, this can be
attributed to the fact that the only low value of L was
used in a very high modulus material. The value of
compression efficiency can be shown by substituting in the
formula of L the value of 120 in the original formula.
In this case, the efficiency drops to 1.125. It may be
seen that a compression efficiency of 1.125 is not a high
efficiency. This value of 1.125 is not a high
value for a member having a diameter of 1.5 inches.

VIII. CONCLUSIONS

The combustion chamber resulting from this design shows a total pressure drop of 2.195 lb/in^2 . G. Geoffrey Smith in Reference 9 states that several standard turbo-jet engines have combustion chambers which operate with a total pressure drop of about 2 lb/in^2 . From this, it would seem that the pressure drop for the designed combustion chamber was not excessive.

The British use pressure drop divided by reference velocity head ($\Delta P/q$) as a parameter for comparing different combustion chambers. Values of this parameter from 20 to 30 are considered reasonable. Here again, the designed combustion chamber conforms, having a $\Delta P/q$ of 21.1.

Another parameter is pressure drop divided by total inlet pressure ($\Delta P/P_{4t}$). Most values of this parameter quoted for standard combustion chambers varied from three to six per cent. However, these values were invariably for total inlet pressures of 3.5 to 4.5 atmospheres. A test of a DeHavilland H-1 combustion chamber, operating at an inlet pressure of 1.8 atmospheres (a value very near the design point of 1.6 atmospheres), showed a $\Delta P/P_{4t}$ of 18%. Thus, it would appear that the value of 9.34% calculated for the designed chamber is acceptable.

Hawthorne's empirical formula for combustion efficiency was given with no indication as to its valid range.

III. CONCLUSIONS

The conditions under which the tests were carried out are given in Table I. It is seen that the pressure drop for the engine was not negligible.

The engine was operated at a constant speed of 1500 rev/min. The pressure drop for the engine was not negligible. The pressure drop for the engine was not negligible.

Further experiments in pressure drop showed that the pressure drop for the engine was not negligible. The pressure drop for the engine was not negligible.

It is concluded that the pressure drop for the engine was not negligible. The pressure drop for the engine was not negligible.

It may be that using this formula under a condition of low inlet pressure is not justified and that actual tests of the chamber would prove the value of 70.95% obtained from it to be much too low a value of combustion efficiency. This would appear to be the case, judging by the results obtained in the aforementioned test of the DeHavilland chamber. Operating under conditions very close to the design point used in this investigation, an efficiency of 94% was obtained.

It is regrettable that time did not permit the building and testing of the design. Only with the test results in hand can it be said positively that the design is good or bad. However, by comparison with proved designs, it would appear to be a satisfactory combustion chamber.

It may be said that this formula makes a condition of the
first element is not justified and that second part of
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it is not for a value of condition at all.
This would appear to be the case. Indeed by the
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will appear to be a satisfactory condition.

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in fact and it is said, namely that the value is
or part. However, no comparison with the value, it
will appear to be a satisfactory condition.

APPENDIX A
OBSERVATION OF FLOW THROUGH SECONDARY HOLES
OF TURBO-JET COMBUSTION CHAMBER

Summary

In order to observe the angularity of flow through the secondary holes of a burner basket, a test box was constructed with entrance, exit, and hole areas similar to those in the combustion chamber of a J-33 turbo-jet engine. The J-33 chamber was selected since it was found to be the most nearly similar to the burner designed in this investigation.

Tufts were suspended from the centers of the holes, and the angle of flow was taken as the angle measured between the tufts and the plate containing the holes during passage of an air stream through the test box.

An attempt to read the flow angle by introducing smoke into the air stream was a failure due to the high velocity and the large amount of turbulence present.

It was concluded that, while the angle of flow varied with distance from the fuel nozzle, an average angle of 45° could be used in determining the area of the secondary holes.

Introduction

If the flow through a hole is at an angle other than 90° , the area allowing fluid to pass will not be the ob-

APPENDIX A

DESCRIPTION OF THE INVESTIGATION

1. PURPOSE AND SCOPE

1.1. OBJECTIVE

In order to determine the possibility of this through the secondary nature of a primary source, a test was made. The test was made with a primary source, with a test which was made in the presence of a 1-10 degree of accuracy. The 1-10 degree was determined as the test was made. The test was made in the presence of a 1-10 degree of accuracy. The test was made in the presence of a 1-10 degree of accuracy.

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1.2. METHOD

The test was made in the presence of a 1-10 degree of accuracy. The test was made in the presence of a 1-10 degree of accuracy. The test was made in the presence of a 1-10 degree of accuracy. The test was made in the presence of a 1-10 degree of accuracy. The test was made in the presence of a 1-10 degree of accuracy.

served area of the hole but the projection of the area on the plane perpendicular to the flow. The formula for flow through an orifice is derived for a flow perpendicular to the orifice. When the flow is not perpendicular to the orifice, the projected area must be used in order for the orifice formula to be valid.

The projected area is

$$A = A_h \sin \Theta$$

where A = projected area

A_h = area of hole

Θ = angle of flow

In this paper, $\sin \Theta$ is called C_p , effectiveness coefficient.

Equipment

Figures 8 and 9 show general views of the equipment used in the experiment. In these figures, the parts are numbered as follows:

- (1) Eight-inch pipe carrying the flow from the rotary compressor located in the basement.
- (2) Water manometer connected to pitot-static tube in the pipe.
- (3) Compressed air line.
- (4) Smoke generator.
- (5) Test box.

Figure 10 shows a close-up of the smoke generator.

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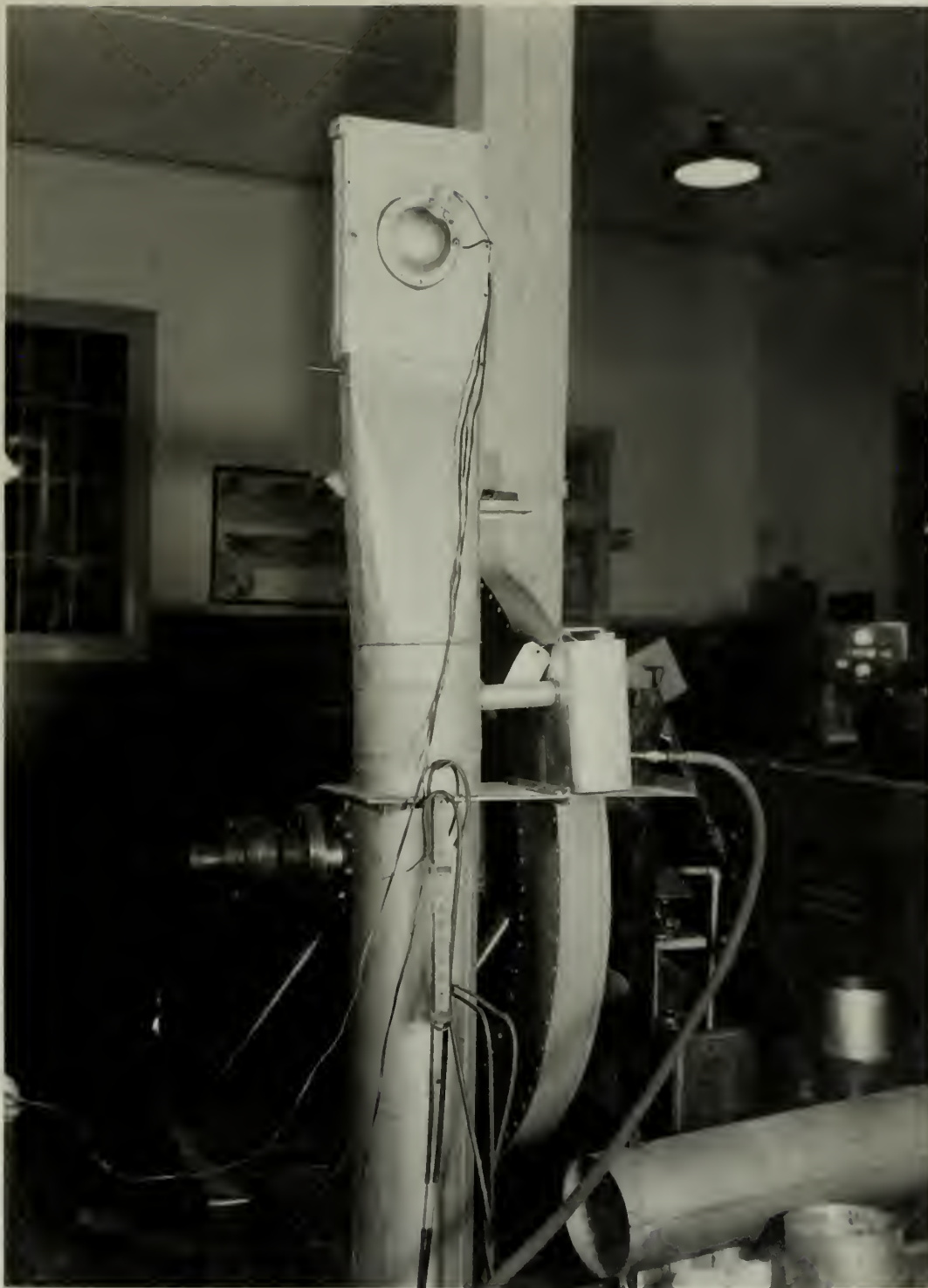


Figure 8. Photograph of Test Setup for Observing Angle of Flow.



Figure 9. Another View of the Test Setup for Observing Angle of Flow.

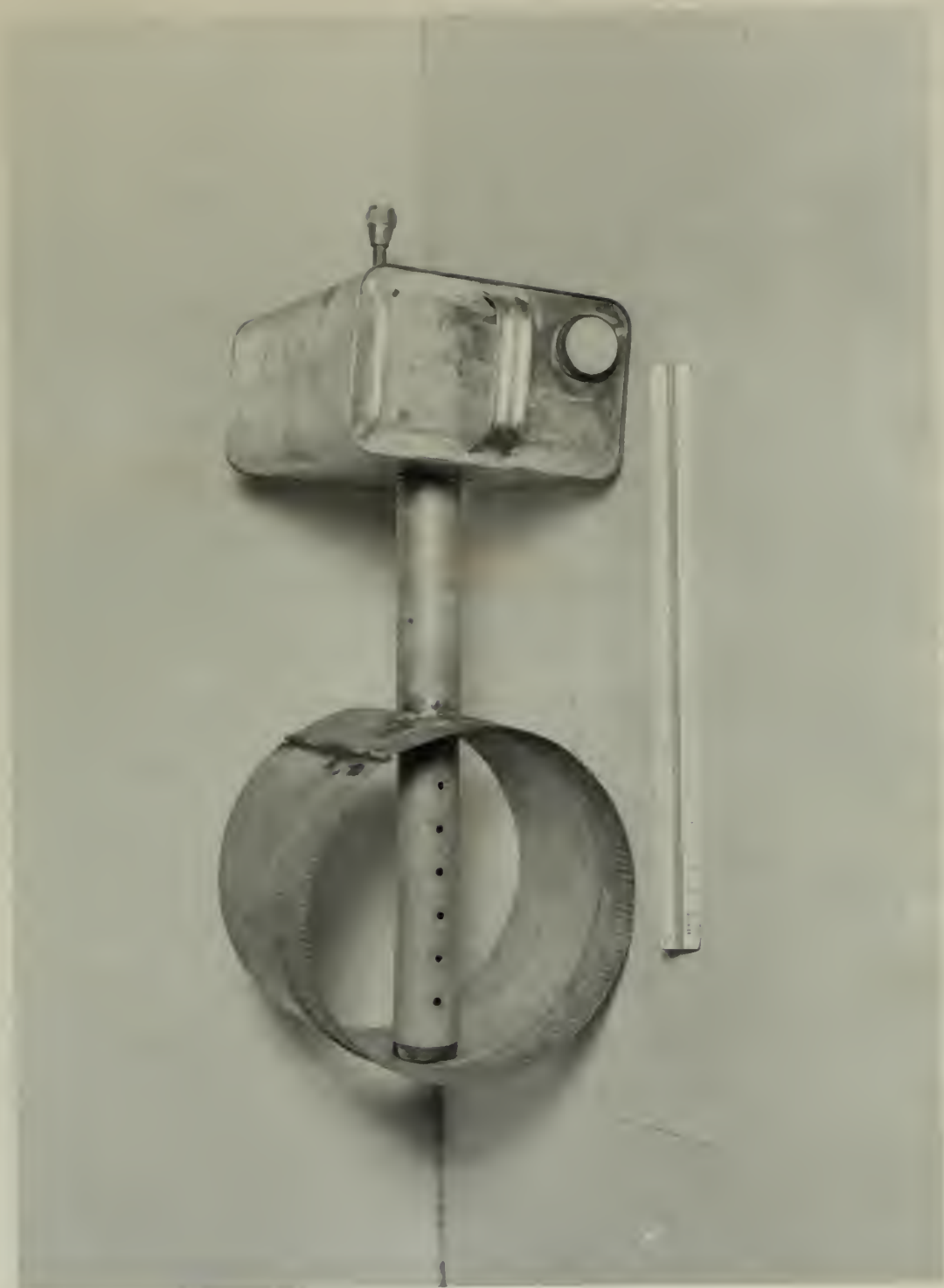


Figure 10. Close-up View of Smoke Generator.

Titanium tetra-chloride ($TiCl_4$) was placed in the gallon can to produce smoke. The amount of smoke was regulated by a valve in the high pressure air line.

Figure 11, a close-up of the test box, shows the window and the lights used in taking pictures of the flow. The test box itself was designed to simulate conditions in a J-33 burner. Its length was the same as the length of the J-33 burner basket. The plate with the holes in it was placed in such a way that it divided the inlet of the box into two areas: one area equal to the clearance area between the outer shell and the burner basket and the other area equal to the area of the holes around the fuel nozzle plus the area of the annular space between the nose hemisphere and the burner basket. In this manner, approximately correct proportions of the flow were introduced on each side of the plate. At the exit to the test box, the plate again divided the flow. The small area at the exit was made equal to the area between the inner and outer walls of the tail cone. The plate was made with the same size and number of holes as the burner basket.

Fine wire was glued across representative holes in the plate and silk thread tied loosely to it. These silk tufts aligned themselves with the flow, making it possible to measure the flow angle.

Procedure

The compressor was started, and, when the flow had

14-00000 (T-14) was placed in the center
and the balance was zeroed. The amount of water was regulated
by a valve in the side pressure air line.

There is a close-up of the last box, where the

The last box itself was designed to resemble something like
the 6-10 pattern. The length was the same as the length of
the 6-10 pattern. The plate with the hole in it
was placed in such a way that it divided the space at the
top into two parts - one even equal to the distance from
between the outer shell and the inner body and the
other also equal to the area of the holes around the hole.
Notice how the area of the smaller space between the two
sections and the lower part. In this manner, apparently,
nearly correct positions of the lips were obtained on
each side of the plate. At the exit on the left hand, the
plate again divides the flow. The wall near the exit
was made equal to the area between the lip and the
wall of the tail cone. The plate was made with its own

and without it joined to the lower part.

These ideas can give more representative value to the time and effort devoted to it. These ideas have raised themselves with the time, and it is possible to answer the time.

Copyright

The experiment was started, and when the 12th day



Figure 11. Close-up View of Test Box.

steadied, readings were taken of the temperature and of the pressure measured by the pitot-static manometer. Then the angle of the tufts was recorded. This was done for three different flows. Photographic records were taken of all runs.

The smoke generator was operated as follows: About 5 cc of $TiCl_4$ were put in the gallon can before the run was started. When smoke was desired, the valve in the high pressure air line was opened. Varying the amount of high pressure air very effectively regulated the amount of smoke obtained. This arrangement gave a very satisfactory supply of smoke under good control. Unfortunately, it was impossible to detect the angle of flow through the holes by this means due to the high velocities and large amount of turbulence.

Results

Table II and Figure 12 show the results of the tests. Photographs taken during the runs are shown in Figures 13, 14, and 15. From the results it can be seen that the flow angle varies from 60° to 35° as the distance downstream increases. There is little variation with change in velocity. The average flow angle over the length of the secondary zone was approximately 45° .

standard, readings were taken at the beginning and at the
 pressure reported by the pilot-observer. The
 angle of the tube was recorded. This was done for every
 different line. The following results were obtained at all
 times.

The angle reported was reported as follows. When
 it was of 10° or more the angle was taken as 10° and
 the other. When the angle was 10° or more the angle was
 reported as 10° and the other. When the angle was 10° or
 pressure at very slightly reported the angle of 10° or
 obtained. This was obtained as a very slight angle of 10° or
 of more than 10° or more. When the angle was 10° or
 angle to obtain the angle of 10° or more the angle of 10° or
 angle was in the 10° or more and large amount of 10° or
 obtained.

Results

Table II and Figure 1 show the results at the
 foot. The following table shows the results at the
 Figure II, 14, and 15. The results at the foot
 and the 10° angle were from 60° to 10° at the 10° or more
 between the results. There is little variation in
 change in velocity. The average 10° angle over the length
 of the roadway was approximately 40°.

Conclusions

Since the average flow angle is 45° , it is apparent that an effectiveness coefficient is necessary for calculating the flow through the secondary holes. In this analytical design, it seemed reasonable to use the average value of flow angle of 45° over the entire length of the secondary zone. Thus, the effectiveness coefficient is

$$C_e = \sin 45^\circ$$

$$= .707$$

Results

When the average flow angle is 45° , it is apparent that an efficiency coefficient is necessary for calculating the flow through the secondary holes. In this design, it seems reasonable to use the average value of flow angle of 45° over the whole length of the main pipe run. Thus, the efficiency coefficient is

$$C_p = 0.85$$

$$C_p = 0.85$$

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Table II. Flow Angle Through Holes for
Various Weight Flows.

Hole No.*	Flow angle - Degrees		
	Weight Flow - lb/sec		
	1.18	1.32	1.45
1	53	62	60
2	45	52	49
3	45	46	45
4	45	45	45
5	42	42	42
6	40	40	40
7	40	37	38
8	38	35	37
9	37	35	35

* Hole number 1 is the first 5/8-inch hole in the test plate. It is located four inches from the entrance to the test box. The successive holes are one inch apart.

Table 11. Flow maps through nodes for
various input flows.

Node No.	Flow maps - 10000 10000 10000 10000	
	10000	10000
1	10000	10000
2	10000	10000
3	10000	10000
4	10000	10000
5	10000	10000
6	10000	10000
7	10000	10000
8	10000	10000
9	10000	10000
10	10000	10000

* While number 1 is the first 10000 flow
in the first node. It is repeated 10000
times from the source to the last node.
The maximum value was 10000.

FIGURE 12

FLOW ANGLE THROUGH HOLES FOR VARIOUS WEIGHT FLOPPERS

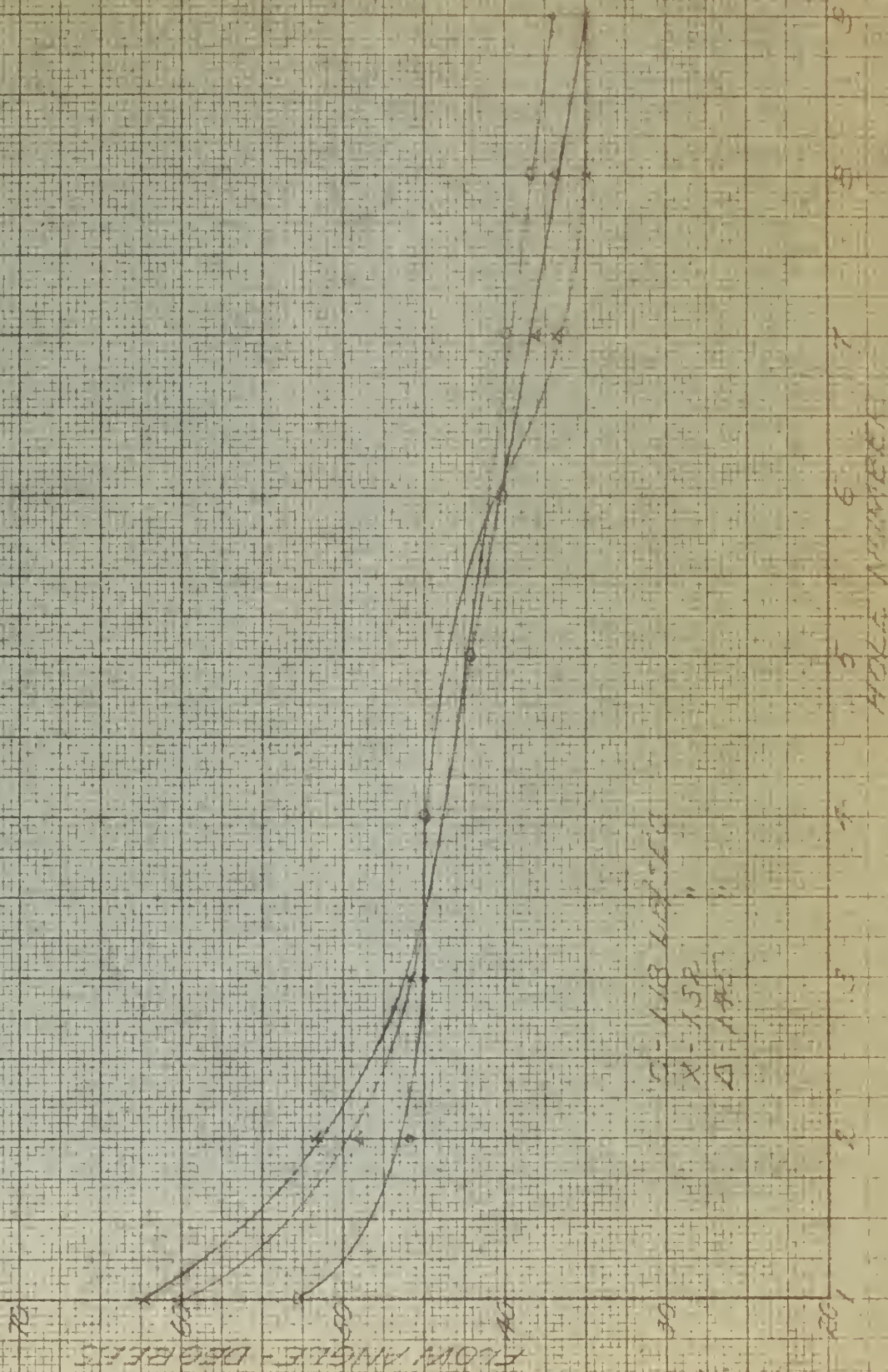




Figure 13. Interior of Test Box Showing Angle of Silk Tufts During Run at 1.18 Lb/Sec.



Figure 14. Interior of Test Box Showing Angle of Silk Tufts During
Run at 1.45 Lb/Sec.



Figure 15. Interior of Test Box During Run in Which Smoke was Used.

Sample Calculations

To find the weight flow through the test box when conditions in the pipe are:

$$\Delta h = \text{pitot-static manometer reading} = 1/2" \text{ H}_2\text{O} = 2.6 \text{ lb/ft}^2$$

$$T = \text{Temperature} = 1300^\circ \text{ F or } 590^\circ \text{ R}$$

$$w = \rho A v$$

where w = weight flow in lb/sec

ρ = density in lb/ft³

A = area of the eight-inch pipe in ft²

v = velocity in ft/sec

$$\rho = .0765 \times \frac{590}{590} = .0675 \text{ lb/ft}^3$$

$A = .785 \times \frac{8^2}{144} = .35 \text{ ft}^2$

$$v = \sqrt{\frac{2g \Delta h}{\rho}} = \sqrt{\frac{64.4 \times 2.6}{.0675}}$$

$v = 49.9 \text{ ft/sec}$

$$w = \rho A v = .0675 \times .35 \times 49.9$$

$$w = 1.18 \text{ lb/sec}$$

$$w = 1.18 \text{ lb/sec}$$

$$\text{Then, } w = .0675 \times .35 \times 49.9$$

$$= 1.18 \text{ lb/sec}$$

w = weight flow through the test box

A = area of the eight-inch pipe

v = velocity of the gas

General Relations

To find the velocity of the gas in the pipe we use

the following relations in the pipe:

$$\Delta p = \text{hydrostatic pressure} = \rho \cdot g \cdot h = 1.2 \cdot 9.81 \cdot 1.5 = 17.65 \text{ Pa}$$

$$\text{Temperature} = 1500 \text{ K or } 1227^\circ \text{C}$$

$$\rho = 1.2 \text{ kg/m}^3$$

$$\text{Area} = \text{cross-section area} = 0.001 \text{ m}^2$$

$$\rho = \text{density of the gas} = 1.2 \text{ kg/m}^3$$

$$\Delta p = \text{pressure difference} = 17.65 \text{ Pa}$$

$$\rho = \text{density of the gas} = 1.2 \text{ kg/m}^3$$

$$\Delta p = \text{hydrostatic pressure} = \rho \cdot g \cdot h = 1.2 \cdot 9.81 \cdot 1.5 = 17.65 \text{ Pa}$$

$$\Delta p = \text{hydrostatic pressure} = \rho \cdot g \cdot h = 1.2 \cdot 9.81 \cdot 1.5 = 17.65 \text{ Pa}$$

$$\Delta p = \text{hydrostatic pressure} = \rho \cdot g \cdot h = 1.2 \cdot 9.81 \cdot 1.5 = 17.65 \text{ Pa}$$

$$\Delta p = \text{hydrostatic pressure} = \rho \cdot g \cdot h = 1.2 \cdot 9.81 \cdot 1.5 = 17.65 \text{ Pa}$$

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$$\Delta p = \text{hydrostatic pressure} = \rho \cdot g \cdot h = 1.2 \cdot 9.81 \cdot 1.5 = 17.65 \text{ Pa}$$

Result

$$\Delta p = \text{hydrostatic pressure} = \rho \cdot g \cdot h = 1.2 \cdot 9.81 \cdot 1.5 = 17.65 \text{ Pa}$$

APPENDIX B

SYMBOLS

- A - maximum area of combustion chamber
- A_1 - clearance area between basket and outer liner
- A_2 - entrance area into basket
- a - speed of sound
- C_e - effectiveness coefficient
- C_o - orifice coefficient
- $C_{p,ave}$ - average specific heat at constant pressure
- C_{p4} - specific heat at constant pressure at inlet
- C_{p5} - specific heat at constant pressure at exit
- D - diameter of basket
- D_1 - diameter of pipe with area equivalent to A_1
- D_h - diameter of each hole in ring
- D_{max} - maximum diameter of combustion chamber
- D_{id} - inside diameter of clearance area
- D_{od} - outside diameter of clearance area
- d - minor diameter of tail cone
- f - friction factor
- g - gravitational constant
- h - length of tail cone
- I - combustion intensity
- L - length from nozzle to turbine entrance
- l - length from nozzle to end of basket
- l_c - length of cylinder or basket

APPENDIX B

SYMBOLS

The symbols used in this report are defined as follows:

α - angle of attack of the airfoil

β - angle of attack of the airfoil

γ - angle of attack of the airfoil

δ - angle of attack of the airfoil

ϵ - angle of attack of the airfoil

ζ - angle of attack of the airfoil

η - angle of attack of the airfoil

θ - angle of attack of the airfoil

ϕ - angle of attack of the airfoil

ψ - angle of attack of the airfoil

ω - angle of attack of the airfoil

χ - angle of attack of the airfoil

ψ - angle of attack of the airfoil

ϕ - angle of attack of the airfoil

θ - angle of attack of the airfoil

ψ - angle of attack of the airfoil

ω - angle of attack of the airfoil

χ - angle of attack of the airfoil

ψ - angle of attack of the airfoil

ϕ - angle of attack of the airfoil

θ - angle of attack of the airfoil

ψ - angle of attack of the airfoil

ω - angle of attack of the airfoil

LHV - lower heating value of fuel
 M_r - reference Mach number
 M_4 - entrance Mach number
 M_5 - exit Mach number
 N - number of combustion chambers
 N_h - number of holes in ring around basket
 NR - Reynolds number
 P - pressure
 P_{sc} - atmospheric pressure at standard conditions
 P_{4s} - entering static pressure
 P_{5s} - exit static pressure
 ΔP - pressure drop
 Q - volume air flow
 q - heat input
 R - gas constant
 T_{sc} - temperature at standard conditions
 T_{4s} - entrance static temperature
 T_{5s} - exit static temperature
 T_{4t} - total inlet temperature
 T_{5t} - total exit temperature
 V - volume available for combustion
 V_C - volume of cylinder or basket
 V_H - volume of hemisphere or dome
 V_{TC} - volume of tail cone
 v - velocity
 v_r - reference velocity

- 100 - lower heating value of fuel
- 101 - reference fuel number
- 102 - reference tank number
- 103 - cold flow number
- 104 - number of combustion chambers
- 105 - number of holes in ring around burner
- 106 - reference number
- 107 - pressure
- 108 - atmospheric pressure at standard conditions
- 109 - entering static pressure
- 110 - exit static pressure
- 111 - pressure drop Δp
- 112 - volume air flow
- 113 - heat input
- 114 - gas constant
- 115 - temperature at standard conditions
- 116 - entrance static temperature
- 117 - exit static temperature
- 118 - total inlet temperature
- 119 - total exit temperature
- 120 - values available for comparison
- 121 - volume of cylinder at bottom
- 122 - volume of cylinder at top
- 123 - volume of cell used
- 124 - velocity
- 125 - reference velocity

v_4 - entrance velocity

v_5 - exit velocity

w - weight flow of air

w_f - weight flow of fuel

γ - ratio of specific heats

η - combustion efficiency

μ - viscosity of air

μ_r - reference viscosity of air

ρ - density of air

ρ_{so} - density of air at standard conditions

- 1 - velocity of air
- 2 - exit velocity
- 3 - weight flow of air
- 4 - weight flow of fuel
- 5 - ratio of specific heats
- 6 - combustion efficiency
- 7 - viscosity of air
- 8 - volumetric efficiency of air
- 9 - density of air
- 10 - density of air at standard conditions

The following are the definitions of the terms used in the preceding list:

1 - velocity of air: The velocity of the air entering the engine.

2 - exit velocity: The velocity of the air leaving the engine.

3 - weight flow of air: The weight of air entering the engine per unit time.

4 - weight flow of fuel: The weight of fuel entering the engine per unit time.

5 - ratio of specific heats: The ratio of the specific heat at constant pressure to the specific heat at constant volume.

6 - combustion efficiency: The ratio of the actual heat released to the theoretical heat released.

7 - viscosity of air: The property of a fluid that resists the motion of one layer of fluid relative to another.

8 - volumetric efficiency of air: The ratio of the actual volume of air entering the engine to the theoretical volume of air entering the engine.

9 - density of air: The mass of air per unit volume.

10 - density of air at standard conditions: The density of air at standard temperature and pressure.

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